JOURNAL of the AMERICAN SOCIETY *of* *NAVAL ENGINEERS* Inc.

November, 1958

Volume 70 Number 4

The ARTHUR G. FESSENDEN *Issue*

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SEVENTIETH YEAR

VOLUME 70 November, 1958 NUMBER 4

Published Quarterly at Washington, D.C., by The American Society of Naval Engineers, Inc.

Entered as Second Class Matter at the Post Office at Washington, D.C., and at Nashville, Tenn. Acceptance for mailing at special rate of postage provided for in Section 1103, Act of October 3, 1917, authorized on July 3, 1918.

Subscription Rates for Non-Members

The subscription price of the JOURNAL, postpaid to points in the United States and possessions is $12.00. Single copies, $3.00. To other countries, $20.00. Single copies, $5.00. All subscriptions payable in advance.

Make checks, drafts and postal money orders payable to THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Advertising rates will be furnished on application.

Addresses will be changed at any time, but only upon notification. Immediate notice should be given of any delay in the receipt of the JOURNAL.

Address all communications for the Society to

THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Suite 1004, Continental Building
1012 14th Street, N.W.
Washington 5, D.C.

The American Society of Naval Engineers, Inc., has no official connection with the Navy Department.

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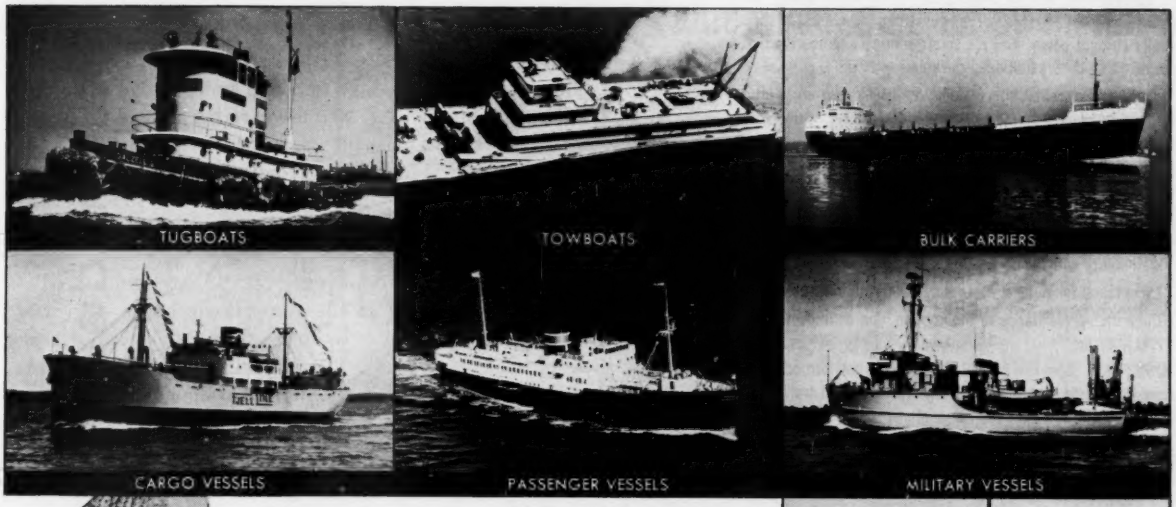

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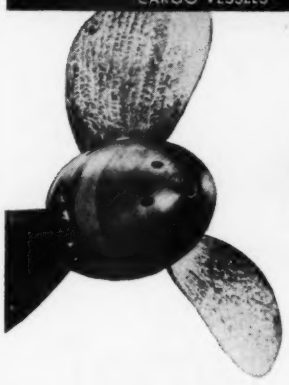

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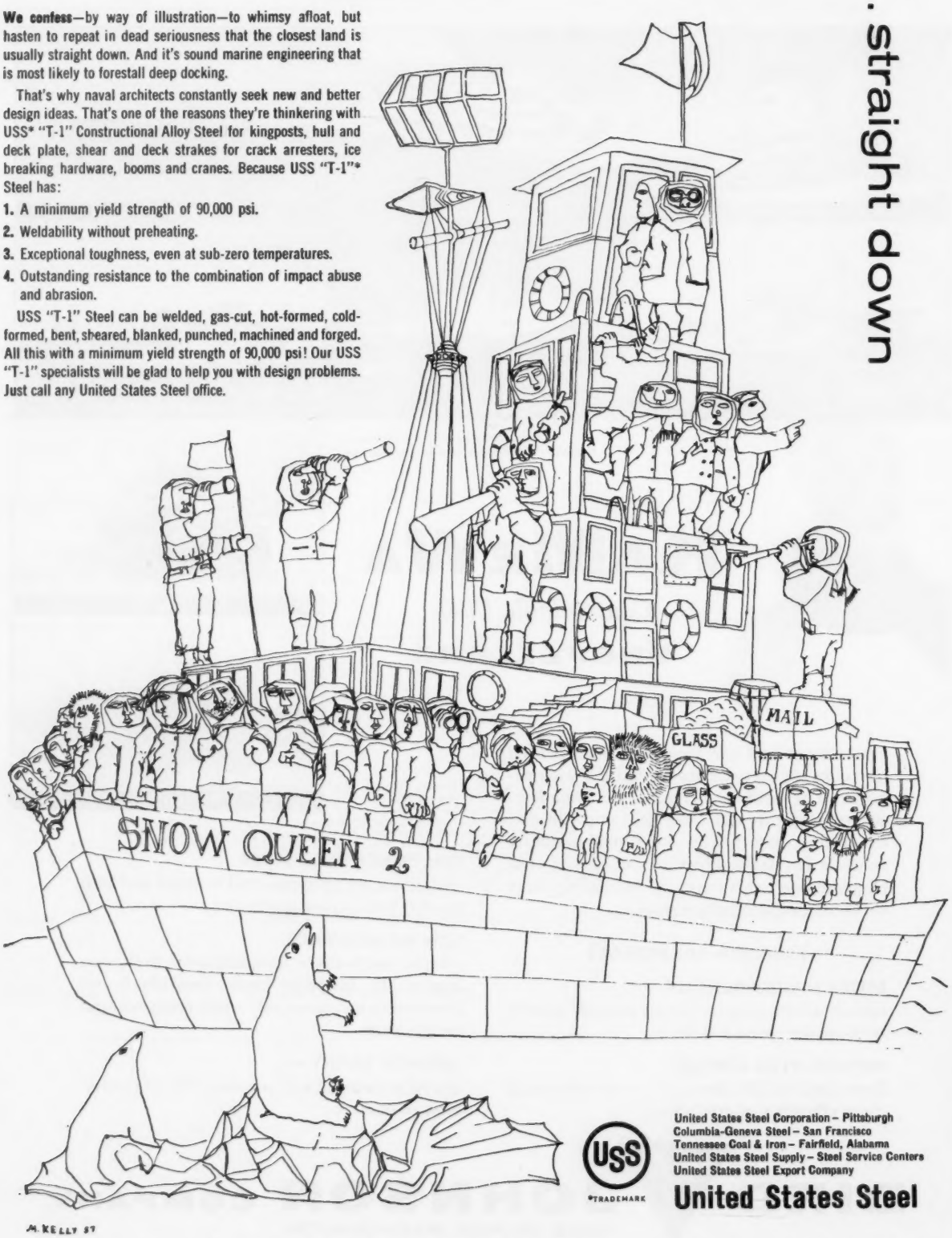

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A reproduction suitable for framing of this highly original drawing by Marie Tuiccillo Kelly is available. Write United States Steel, Room 2801, 525 William Penn Place, Pittsburgh 30, Pa.

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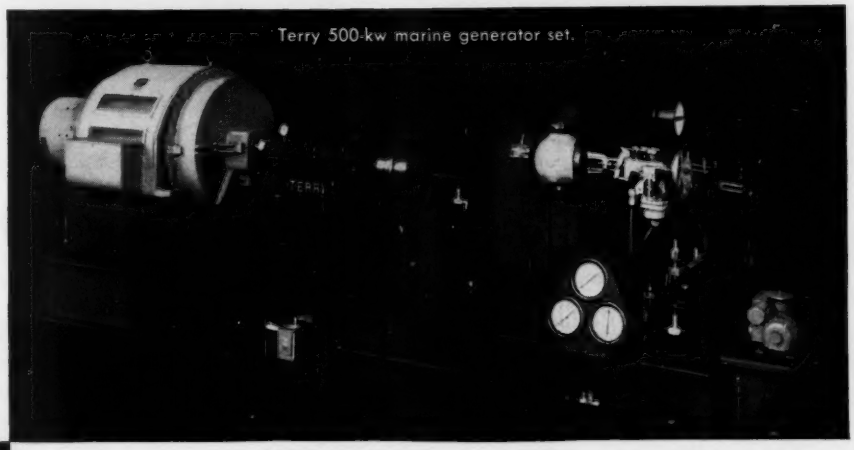
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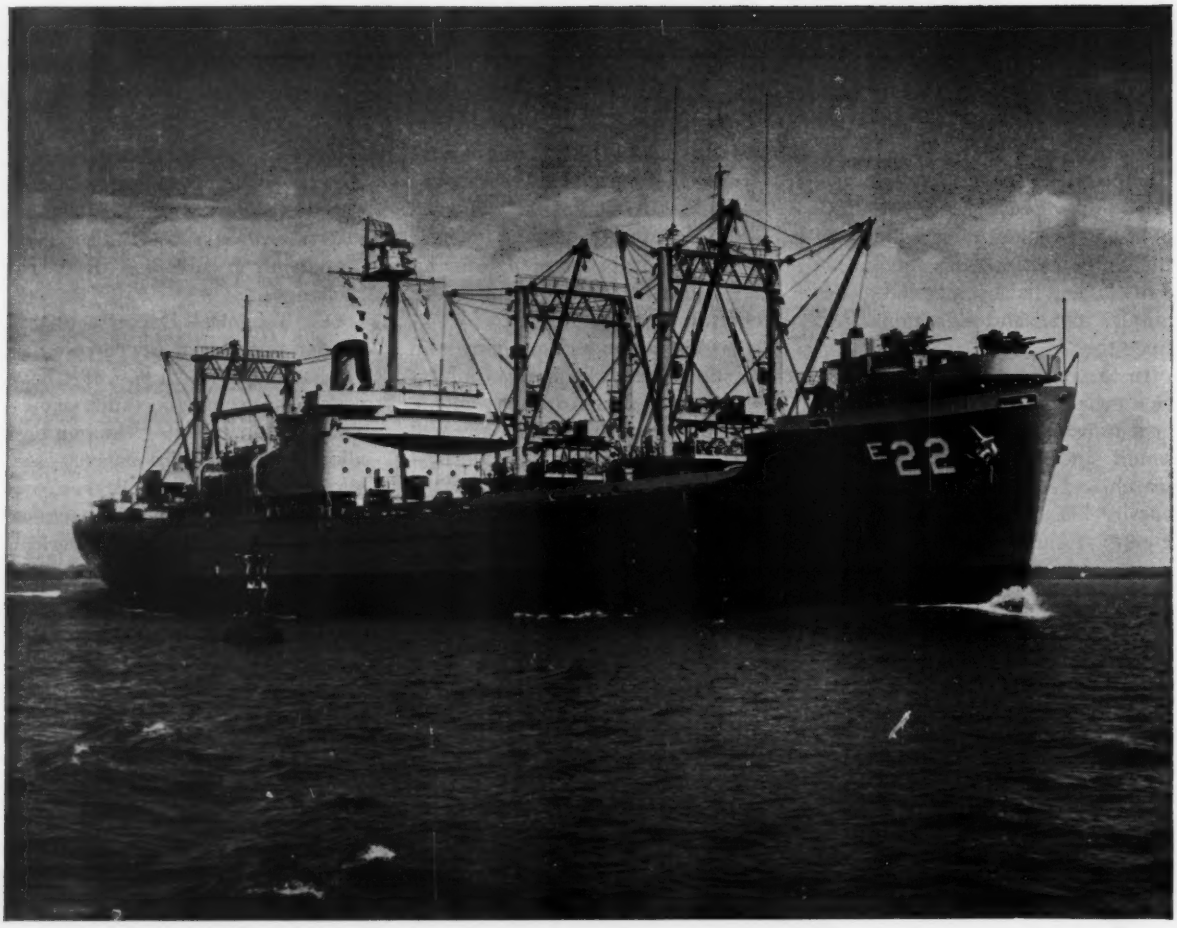

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Cargo Capacity: 7500 tons
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Armament: Four 3 in. 50 Twin Mounts
Complement:
20 Officers, 330 Enlisted Men

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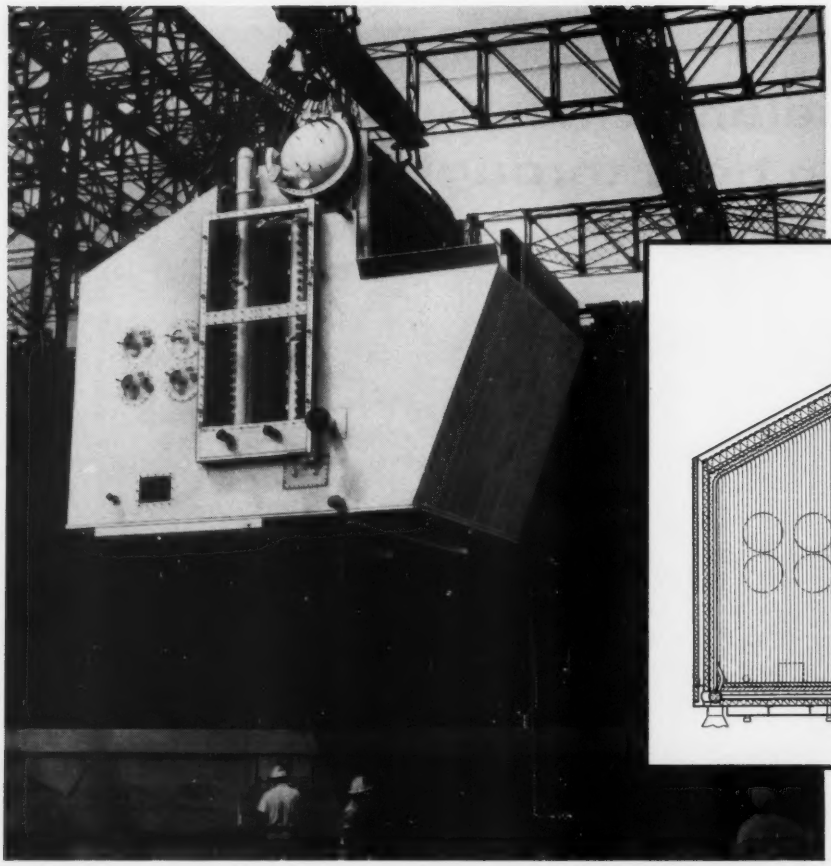
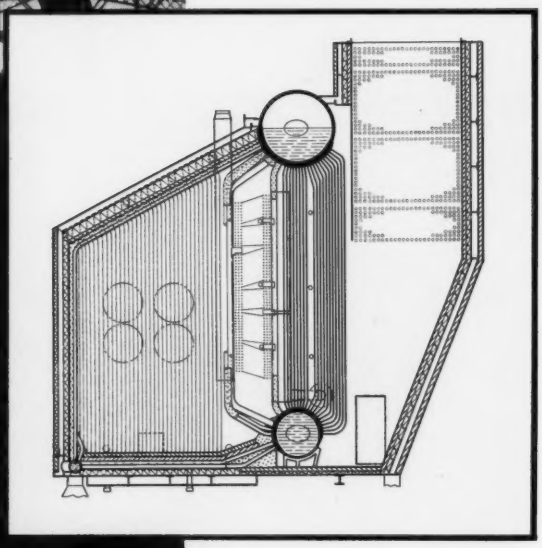


Pre-assembled B&W Two-Drum boiler is swung into position in the "Torrey Canyon". Boiler, diagrammed below, will supply 91,000 lbs of steam per hour per boiler to the 25,000 SHP main propulsion system. Boiler efficiency will be 88% when supplying steam at 850 psi and 860 F total steam temperature.



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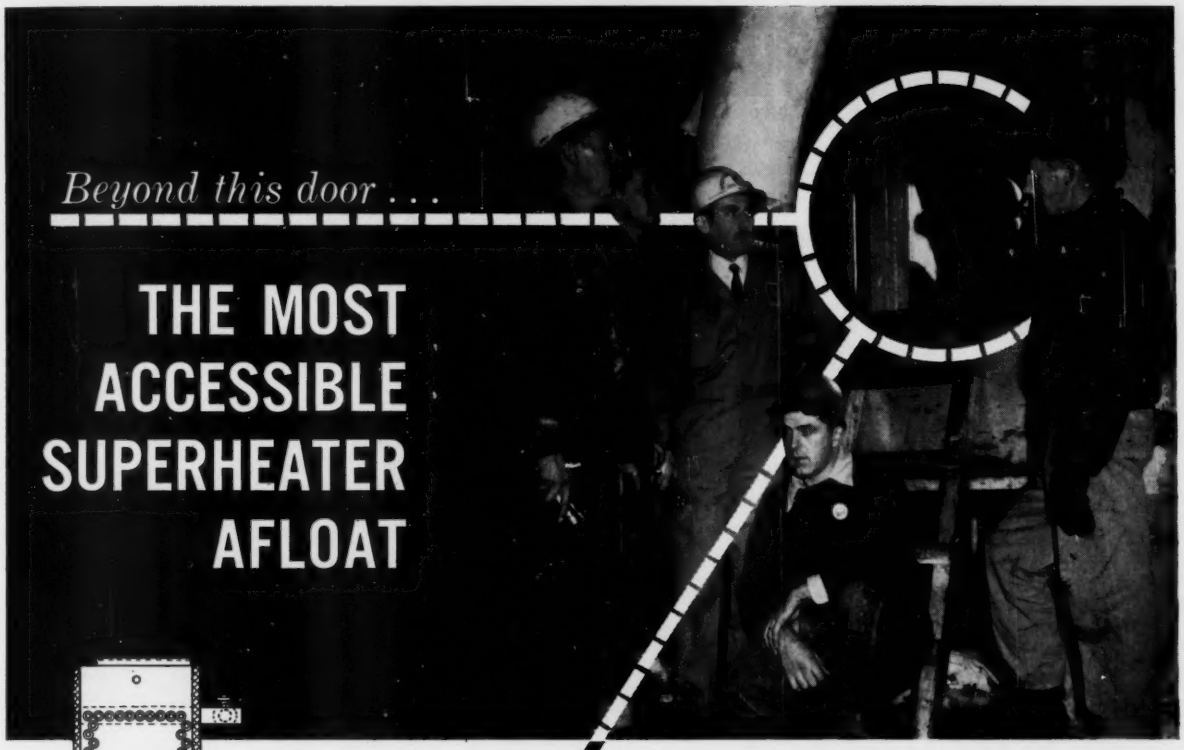
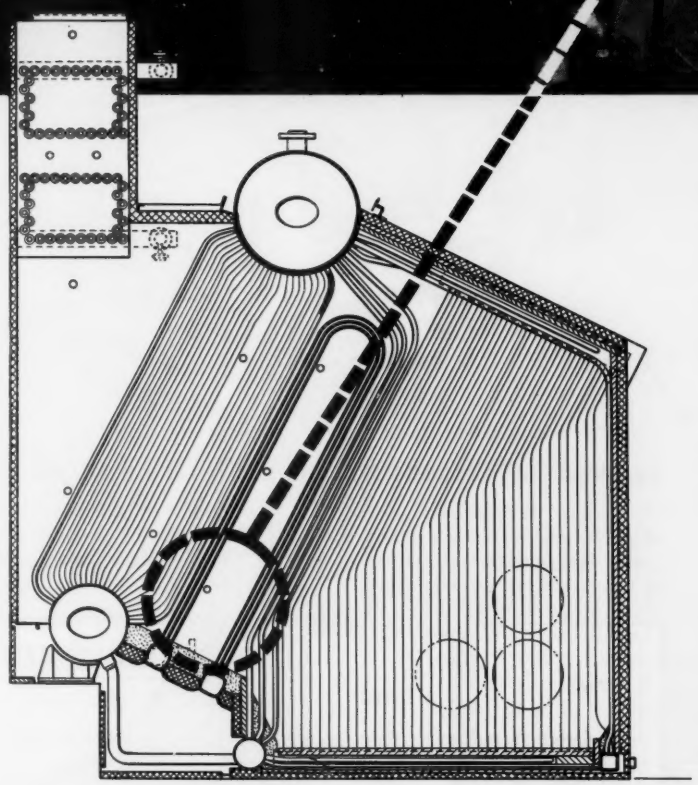


THE MOST ACCESSIBLE SUPERHEATER AFLOAT



This photograph illustrates a vital point about the design of the C-E Vertical Superheater Boiler: the *complete accessibility of the superheater* for inspection and maintenance. The picture shows five men who have just emerged from the superheater space. The sixth is on his way out. Not that you ever need six men in the superheater space, but there *is* room.

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- no horizontal slag-collecting surfaces
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- improved casing design . . . less possibility of leakage

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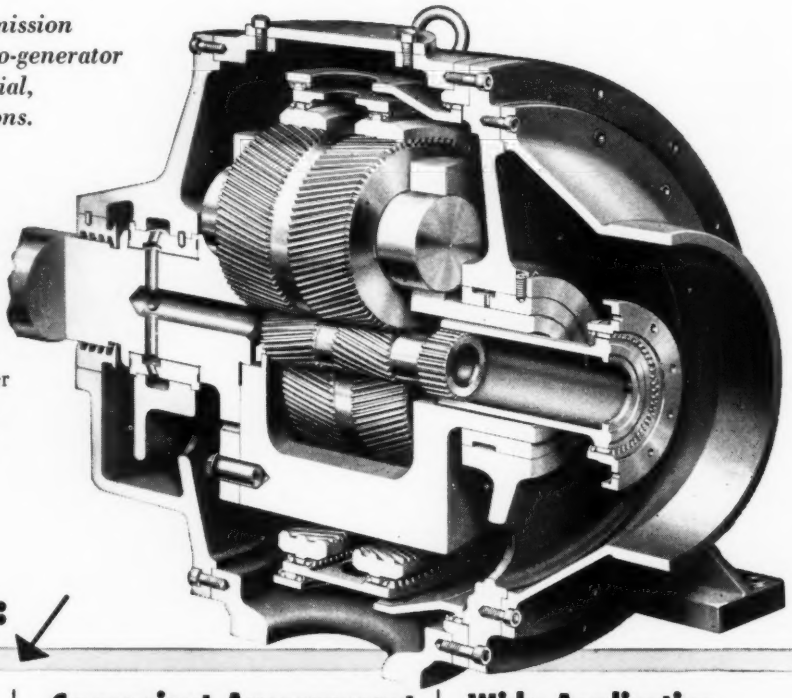
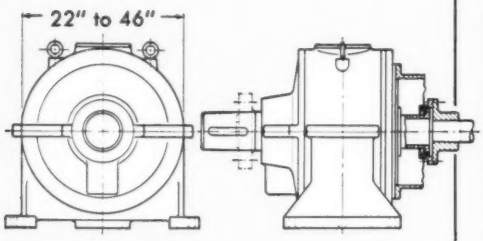
This cutaway view of the De Laval-Stoeckicht Planetary Gear shows how it provides flexibility for proper load distribution throughout the gear members. The thoroughly proved and tested design is completely reliable in transmitting high horsepower for high speed applications. • Highest efficiencies (98% or higher)...no high speed bearings...less friction losses.



Check These Advantages:

Small Size—Light Weight

Compact—low weight per hp. Sizes range from 22" to 46" in diameter, depending on horsepower requirements. Example: 5000 hp planetary unit weighs 1700 lbs. against 6000 lbs. for conventional gear.



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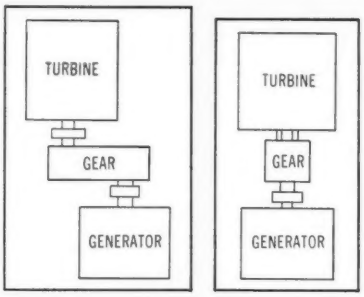

Wide Application

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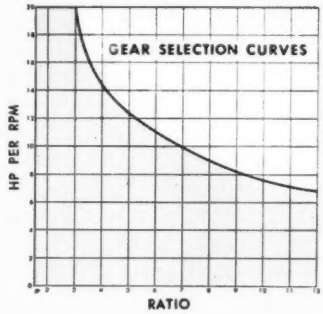
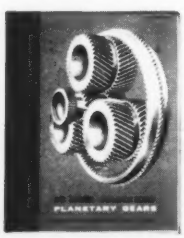




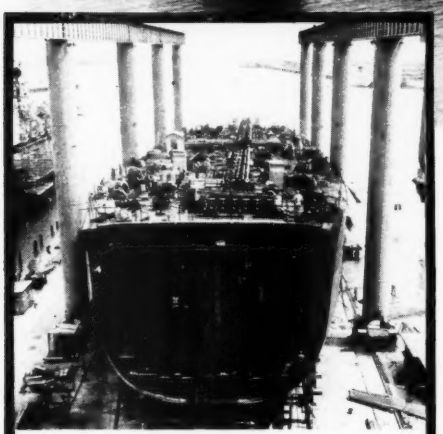
For further details, write for Bulletin 2400.

DE LAVAL *Steam Turbine Company*

Nottingham Way, Trenton 2, New Jersey

THIS NEW SHIP...

in 66 days



Completed 353' Midbody Section constructed at Todd Shipyards, Los Angeles Division, San Pedro, California for the jumboized S.S. DAVID E. DAY

NATIONWIDE...

FOR SPEED
ECONOMY
VERSATILITY

Almost 50 feet longer, 7 feet wider... 25% extra cargo capacity in Jumboized T2 Tanker

Modern techniques, fast delivery are the TODD formula for ship conversions. The S.S. DAVID E. DAY of Richfield Oil Corporation, entered Todd Shipyards on May 26 and was back in service on July 31, 1958 . . . completely jumboized with new prefabricated Midbody Section. Now 572 feet long, this tanker has an increased capacity of about 1,400,000 additional gallons.

At a fraction of the time required to build a new vessel and with new-ship life expectancy, the S.S. DAVID E. DAY incorporates the excellent propulsion system of the T2 tanker plus latest design features . . . from construction with through longitudinals to new cargo system and corrosion control.

TODD-quality workmanship at 8 self-contained shipyards is your guarantee of days saved . . . dollars saved!

TODD SHIPYARDS CORPORATION

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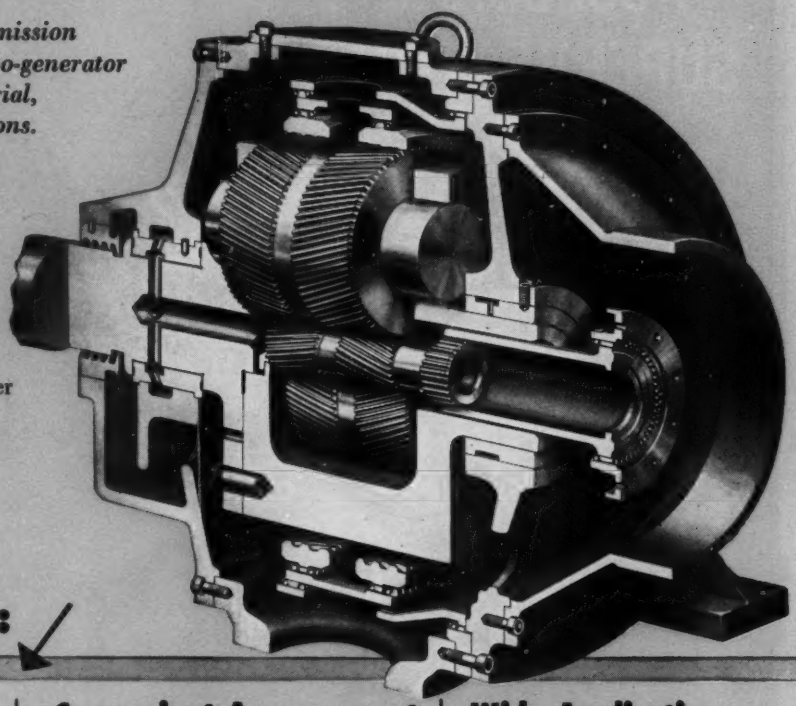
DE LAVAL-STOECKICHT
PLANETARY GEAR

...for high speeds...high horsepower

Proved in hundreds of installations abroad totalling over 3,000,000 horsepower—now available in America!

For all high torque power transmission applications such as pump turbo-generator and compressor drives in industrial, municipal and marine installations.

This cutaway view of the De Laval-Stoeckicht Planetary Gear shows how it provides flexibility for proper load distribution throughout the gear members. The thoroughly proved and tested design is completely reliable in transmitting high horsepower for high speed applications. • Highest efficiencies (98% or higher)...no high speed bearings...less friction losses.



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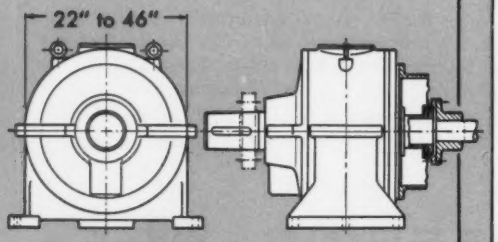

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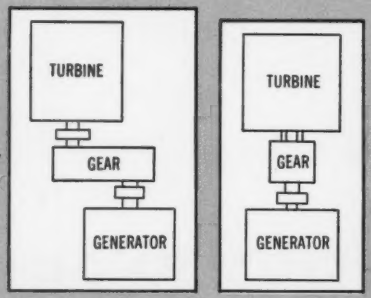

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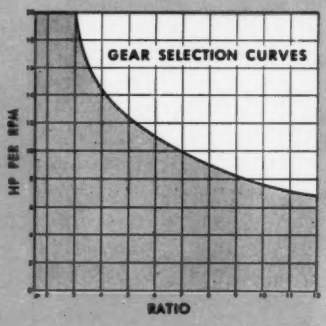

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NORTH POLE SUBS WESTINGHOUSE

USS SKATE

Second to conquer the polar ice cap—just 8 days later! Under Commander James F. Calvert, the Skate passed under the North Pole at 9:47 P. M. (EDT) on August 11. The Skate is also powered by a Westinghouse atomic reactor.

FIRST IN ATOMIC POWER

Six of the first seven atomic submarines—powered by Westinghouse. In addition to the NAUTILUS *and the* SKATE, *they are:*

USS SARGO
Went to sea on August 3, 1958.

USS SWORDFISH
Went to sea on August 9, 1958.

USS SKIPJACK
Launched on May 26, 1958.

USS SEADRAGON
Launched on August 16, 1958.

The atomic reactors for these submarines were developed by Westinghouse under the direction of and in technical cooperation with the Naval Reactors Branch of the U. S. Atomic Energy Commission.

S BOTH POWERED BY ATOMIC ENGINES

USS NAUTILUS

First to pioneer a new trail under the polar ice cap. Under Commander W. R. Anderson, it reached the North Pole at 11:15 P. M. (EDT) on August 3. The Westinghouse atomic reactor, operating on a small quantity of uranium, enabled the Nautilus to make this 8,000 mile trip *without refueling, and almost entirely under water.*

WESTINGHOUSE RECEIVED THIS WIRE FROM COMMANDER ANDERSON OF THE NAUTILUS

CLASS OF SERVICE
This is a full-rate Telegram or Cablegram unless its deferred character is indicated by a suitable symbol above or preceding the address.

WESTERN UNION

W. P. MARSHALL, PRESIDENT

SYMBOLS
DL = Day Letter
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Ship Radiogram

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P WA 625 GOVT NL PD=THE WHITE HOUSE WASHINGTON DC AUG 8 1958
WESTINGHOUSE ELECTRIC CORP=
PGH=

DURING NAUTILUS NORTH POLE SUBMERGED TRANSIT FROM PACIFIC TO ATLANTIC THE PERFORMANCE OF OUR ENGINEERING PLANT EXCEEDED ALL EXPECTATIONS. TO THE FIRST MANUFACTURER OF NAVAL NUCLEAR PROPULSION OUR SINCERE THANKS FOR PROVIDING THE PLANT THAT MADE POSSIBLE THIS FIRST TRANSPOLAR CROSSING=
COMMANDER W R ANDERSON COMMANDING OFFICER
U.S.S. NAUTILUS=

J-92020

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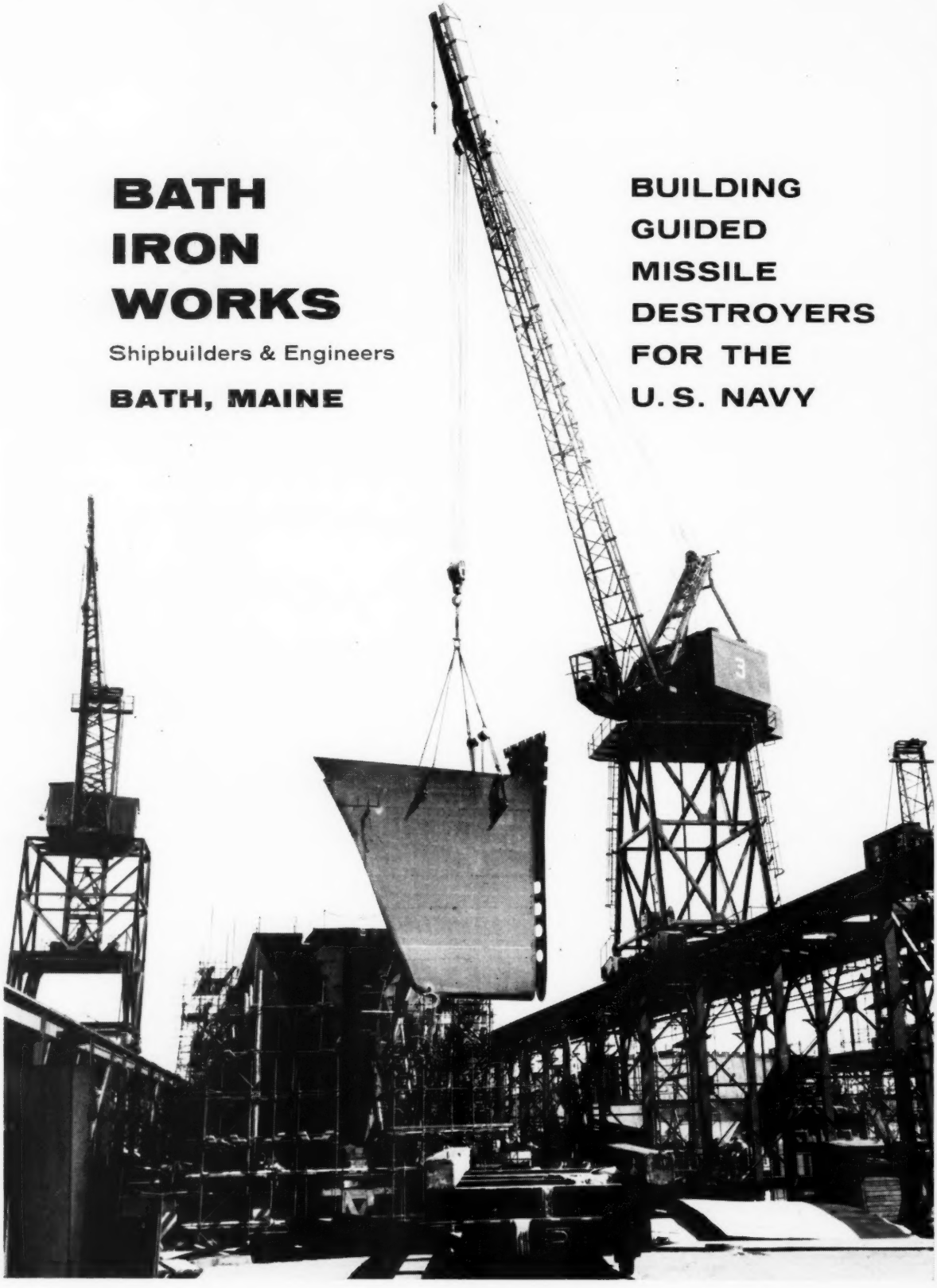
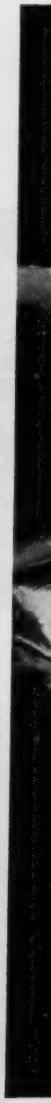
Shipbuilders & Engineers

BATH, MAINE

BUILDING GUIDED MISSILE DESTROYERS FOR THE U. S. NAVY





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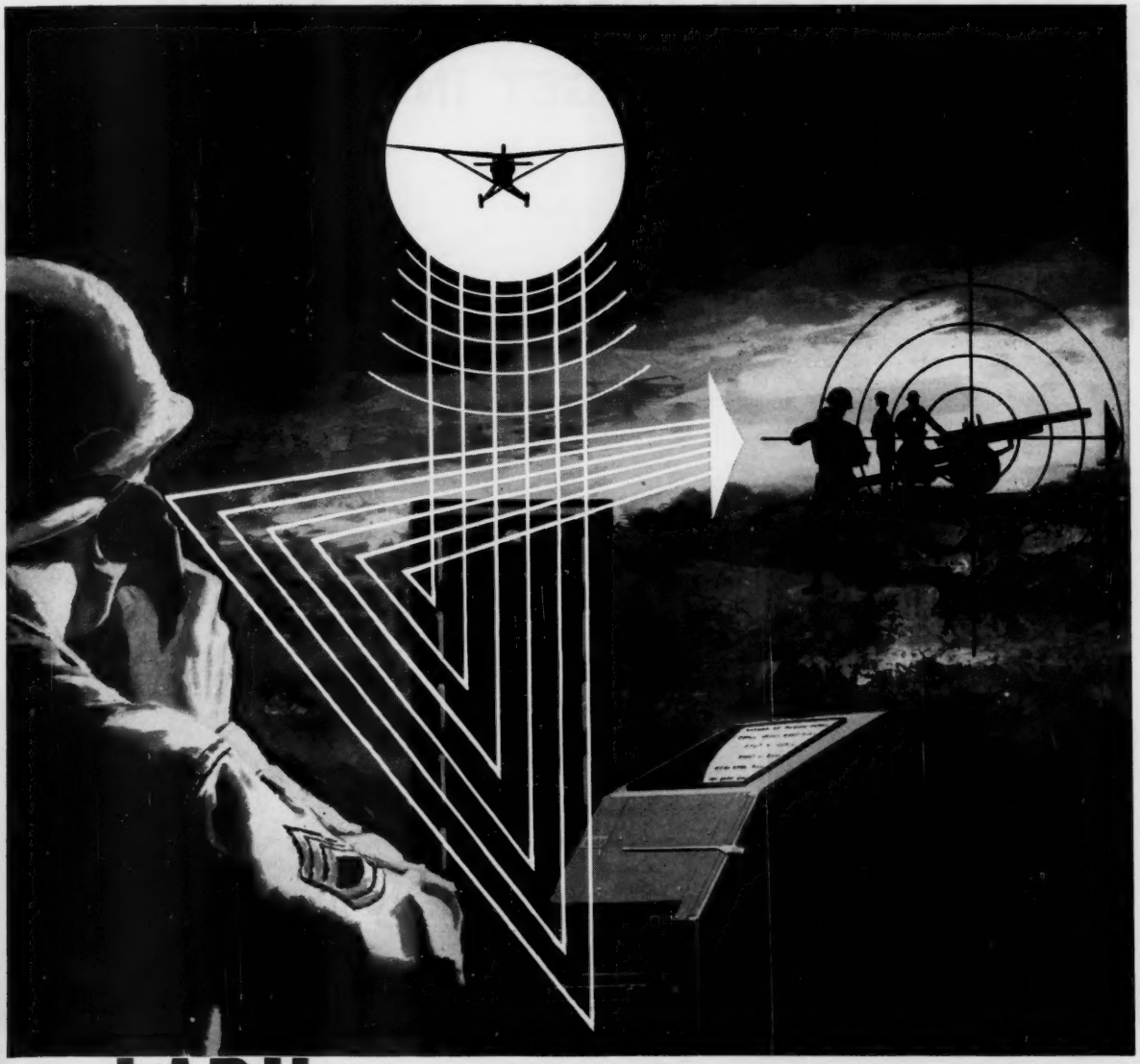

Why LABIL is a powerful link in the communications chain

Speed and reliability make LABIL, Stromberg-Carlson's new data link, ideal for automatic transmission of flight information from light aircraft to ground receiving and control locations.

Into the link the pilot or observer can enter 13 types of data regarding flight and target. When the ground control group wants the information, a lamp on the panel of the airborne equipment lights. The pilot or observer presses the transmit key, and the entire stored message is automatically transmitted over his *existing voice communications equipment.*

Greatly increased reliability is achieved by transmitting each character *twice.*

At the receiving end the message is checked for errors due to noise interference. The error detector examines the two transmissions for complete agreement, then prints the message out on a teletypewriter. Speed of transmission is limited only by the bandwidth of the communications equipment and printout device.

The standard format and digital nature of each transmission make LABIL easily adaptable to large-scale operational control systems in which automatic data handling is a requirement.

Complete technical data on Stromberg-Carlson's Light Aircraft Binary Information Link is available on request.

"There is nothing finer than a Stromberg-Carlson"

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NAVY'S POLARIS MISSILE TO REACH VIRTUALLY ANY TARGET IN WORLD

1500-mile missile will be fired underwater from sub

THE STORY BEHIND THE STORY

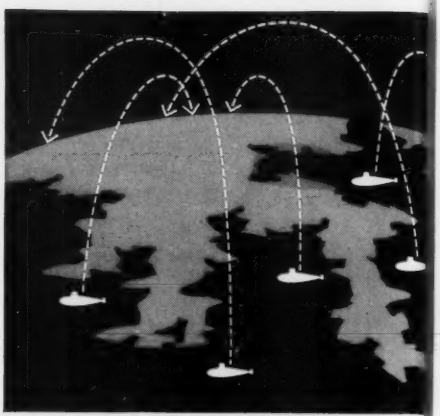
When the new Polaris weapon system becomes operational, it will give our Navy the power to deliver a decisive retaliatory blow at almost any enemy target on earth. Polaris can be launched from an atomic sub hidden in the ocean depths, and its 1500-mile range can bring virtually every major military area of potential aggressors under fire. Defense against Polaris is said to be nearly impossible since the direction of attack cannot be foreseen.

Launching a missile at a distant target from a maneuvering atomic sub presents extraordinary navigation problems. Location of the sub must be known precisely.

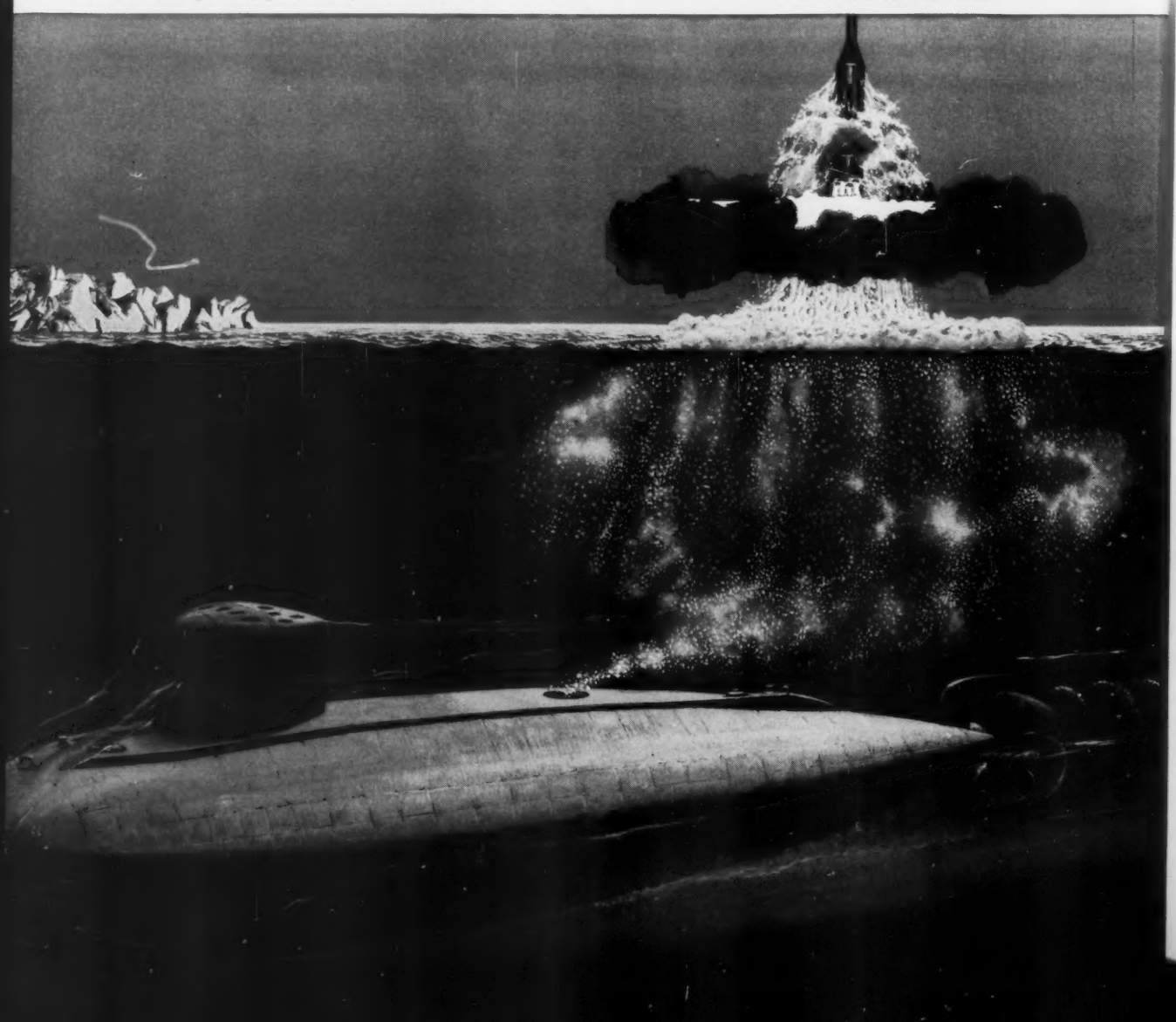
To provide exact navigation data, Sperry is developing for the Navy advanced electronic and gyroscopic systems that will stabilize the sub, continuously establish its precise position and true speed, and feed target data automatically into the missile's guidance system.

Sperry's efforts, joined with those of the missile system manager—Lockheed—and other companies involved, will make Polaris one of the most powerful deterrents in America's strategic defense arsenal.

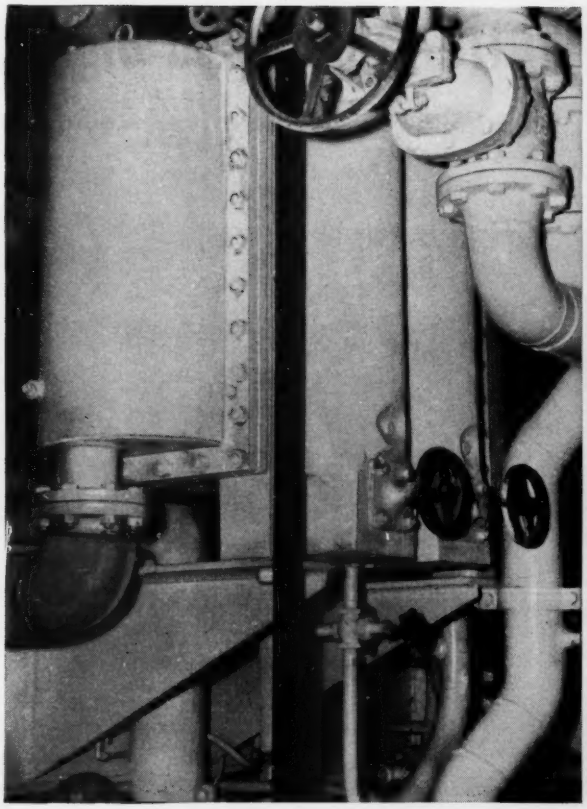
SPERRY GYROSCOPE COMPANY
Great Neck, New York
DIVISION OF SPERRY RAND CORPORATION



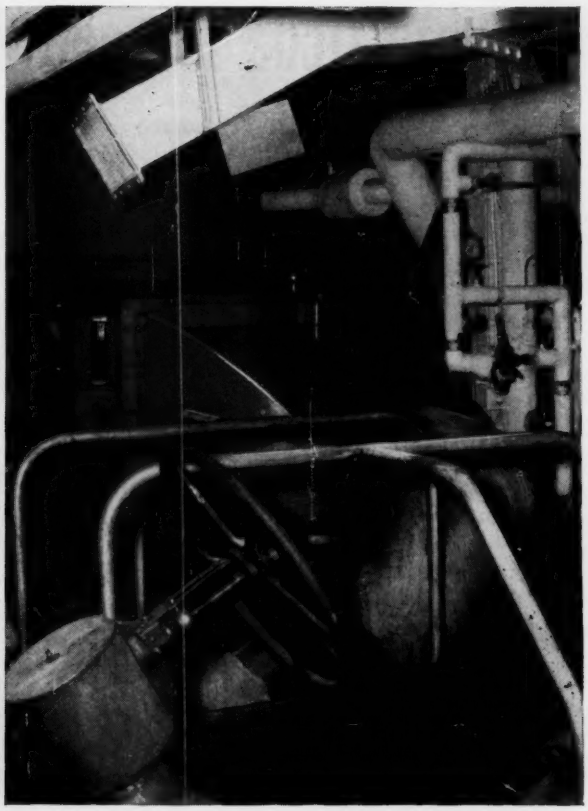
STRATEGIC IMPORTANCE of Polaris is seen in this symbolic map showing possible launching sites. Every major body of water on earth is potential site for Polaris. 1500-mile range covers most of world's land area.



ATOMIC SUBS, guided to exact launching position by Sperry navigation system, launch Polaris missiles from remote areas. Ship's Inertial Navigation System (SINS) requires no contact with shore stations to establish sub's position, speed, course, and true North.



Compact design and ready accessibility of C. H. Wheeler Twin-Element, Two-Stage "Tubejet" Air Ejectors contribute to efficient machinery arrangement in the U.S.N.S. Comet.



Two C. H. Wheeler Condensers, hung from 6000 hp GE turbines, were designed to turbine requirements. They give minimum condensate depression and effective steam distribution.

The first ship of its kind ever built—and she's WHEELER-EQUIPPED!

The U.S.N.S. *Comet* is the first ship ever built specifically for roll/on, roll/off cargo service. As such, she's unique—but in one respect the *Comet* has much in common with other large vessels, new and old.

She's equipped with C. H. Wheeler Main Condensers and "Tubejet"® Air Ejectors. Each Condenser has 4,865 sq. ft. of area, is of the Single Pass type, and is designed for scoop injection. Construction is of ¾" O.D., 18 B.W.G., 90-10 copper-nickel tubes and tube plates. The "Tubejet" Air Ejectors on the *Comet* are of the twin-element, two-stage design with surface inter-after condenser and gland steam condensing sections.

Next time you're planning on marine condensers check on C. H. Wheeler's lightweight, compact design. Find out about "Tubejet" Steam Jet Air Ejectors, with integral piping and valving, too.

Marine Division

C. H. Wheeler Mfg. Co.

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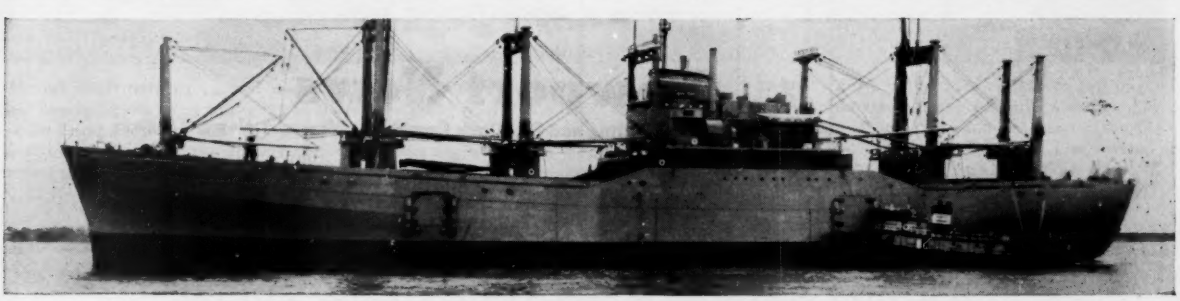
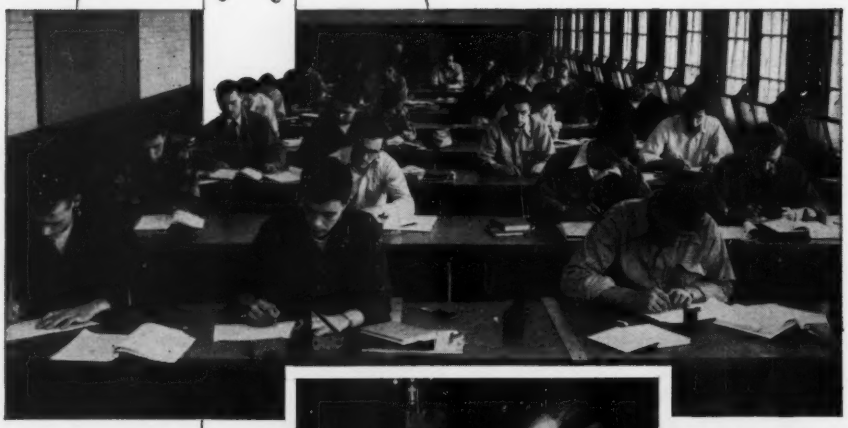
PHILADELPHIA 32, PA.

Marine Condensers, Ejectors, Pumps & Auxiliary Machinery • Steam Condensers • Vacuum Equipment • Centrifugal, Axial & Mixed Flow Pumps • Nuclear Products





Mechanical drawing class of Apprentice School students at Newport News. Those who excel advance to the technical drafting departments.

Shop training scene which shows an apprentice shaping a small die on a shaper, indicates how students are thoroughly trained.

How Skills are born... at Newport News

Newport News utilizes the skills of almost every known trade ...

And for the past three and a half decades, this Company's Apprentice School has furnished substantial numbers of the trained leaders and skilled craftsmen who make outstanding ships the tradition of Newport News.

This school's faculty . . . including experts for both shop practice and academic subjects . . . provides instruction equivalent to a Junior College education during four and five year courses.

Graduates numbering some 2200 skilled workers in 20 separate crafts comprise one more reason why quality identifies everything produced at Newport News.

Newport News

Shipbuilding and Dry Dock Company · ***Newport News, Virginia***

Jungle missionary straightens out shaft problem

Hazardous passage. Drifting debris, submerged rocks and ever-shifting shoals pose constant threats to this missionary. He is Brother Dismas of the Maryknoll Fathers, whose frequent missions take him up and down uncharted, unpredictable waters deep in the jungles of Bolivia.

There's no telling what he may run into here. One of the menaces is the dead *Tajibu* or "vegetable iron" tree.

A giant hardwood, the *Tajibu* dies and topples into the river. The wood is so heavy the whole tree sinks. And so *hard* it's been used in place of bearing metal. Imagine what happens when a *Tajibu,* cartwheeling along in the swiftly running current, tangles with the boat's slender drive shaft . . ! It's virtually impossible to detect one fast enough to avoid being rammed.

Man-eating Piranhas and alligators lurk in these jungle rivers, but Brother Dismas dives in and removes bronze propeller shaft, bent in collision with an unseen *Tajibu.* With nearest help several days' journey away, he has no alternative but to attempt an on-the-spot repair.

Riverside "shop"—The bent bronze shaft is first heated over an open fire, then placed on a log and hammered straight. This primitive repair job has to get him back to his base. "After a few experiences like this, I got a bright idea," says Brother Dismas. "I decided to try one of your Monel shafts."

"Monel shaft arrives"... and was I glad to see it. I've found that bronze shafts bend too easily in these rivers and steel shafts corrode and chew up bearings. We picked up our new Monel shaft in an ox cart at the airport and were underway with it in record time. Now we have...

"A sturdy shaft at last!" For a better idea on how to get longer service from shafts, fuel and water tanks, fastenings and rigging — and a simple explanation of how corrosion can destroy marine equipment, write for "When Metals Go To Sea," The International Nickel Company, Inc., 67 Wall St., New York 5, N. Y.

INCO *Registered trademark

INCO NICKEL ALLOYS

NICKEL ALLOYS PERFORM BETTER LONGER

GIBBS & COX, INC.

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FOR MORE THAN 33 YEARS, BLUDWORTH MARINE has designed and precision-built electronic equipment and navigational aids for the Government and the Shipping Industry, for installation on naval vessels, passenger liners, cargo ships and tankers throughout the world.

ELECTRONIC NAVIGATION DEVICES
RADIO DIRECTION FINDERS
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RADIO TELEPHONES
UNDERWATER TELEVISION CAMERAS
"POWER DIVER"

BLUDWORTH MARINE

Division of KEARFOOT COMPANY, INC.
1500 Main Avenue, Clifton, N.J.

A SUBSIDIARY OF GENERAL PRECISION EQUIPMENT CORP.

KINGSBURY THRUST and JOURNAL BEARINGS

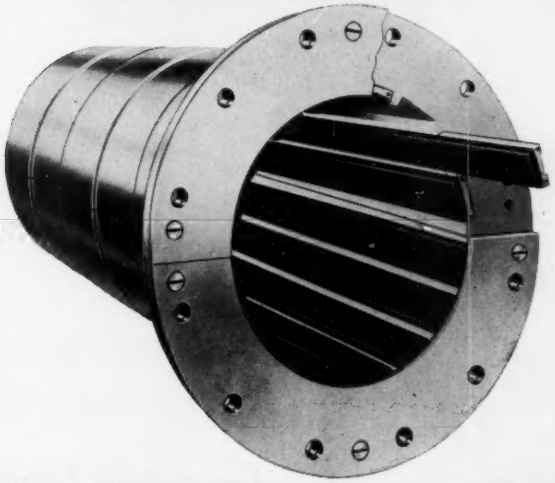
THE NAVY'S CHOICE FOR 40 YEARS

Kingsbury Machine Works, Inc. KINGSBURY Frankford, Philadelphia 24, Pa.

CUTLESS BEARINGS

for

Stern Tubes and Struts



Soft rubber bearing surface—efficiently lubricated by water—this bearing far outlasts all hard surface types, protects propeller shafts, reduces vibration. More than pays for itself in extra wear alone. Saves you time, trouble, and upkeep expense.

LUCIAN Q. MOFFITT, INC.

Akron, Ohio

For Sale:

The original SLOCUM system of scientific propeller design with complete design data for 3, 4 and 5 blades.

Insures best performance with no model tests required.

Reference:

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Incombustible Joiner Materials • Acoustical Materials
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Boiler and Engine Room Insulations • Packings • Gaskets

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BEEKMAN 3-7430

Capacities: ½ to 3 tons

Yale Rail King tractor trolleys with accelerating brake mean

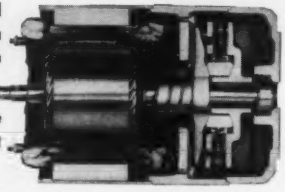
Greater Production, Economy and Safety

Yale Rail King Tractor Trolleys boost production efficiency because they give your hoists greater flexibility of use...can be used with *any* hand or electric hoist with a load capacity up to 3 tons. Your hoists "go" where they are needed...do more on-the-spot lifting jobs...save valuable production time.

Furthermore, Yale Rail Kings—ideal for moving materials in a production line at a steady pace—can pull a string of loaded trolleys easily because a spring-loaded, rubber-tired wheel provides plenty of traction. This wheel rolls against the underside of the track, thus assuring efficient operation on standard I-beams as well as special rolled tracks.

Yale Rail Kings offer you the best in operating and safety features—features that assure years of dependable service and big savings in time, effort and maintenance costs. Your low, initial Rail King investment quickly writes itself off in daily handling savings. For complete information, call or write us.

5 BIG SAFETY FEATURES OF YALE ACCELERATING BRAKE



- Permits inching and positioning of a load with maximum safety.
- Gives adjustable acceleration to suit load requirement.
- Gradual deceleration prevents abrupt stops, thus eliminating danger of swinging loads.
- Positive holding brake assures no drift of load at rest.
- Suitable for plug reversals in the event of an emergency.

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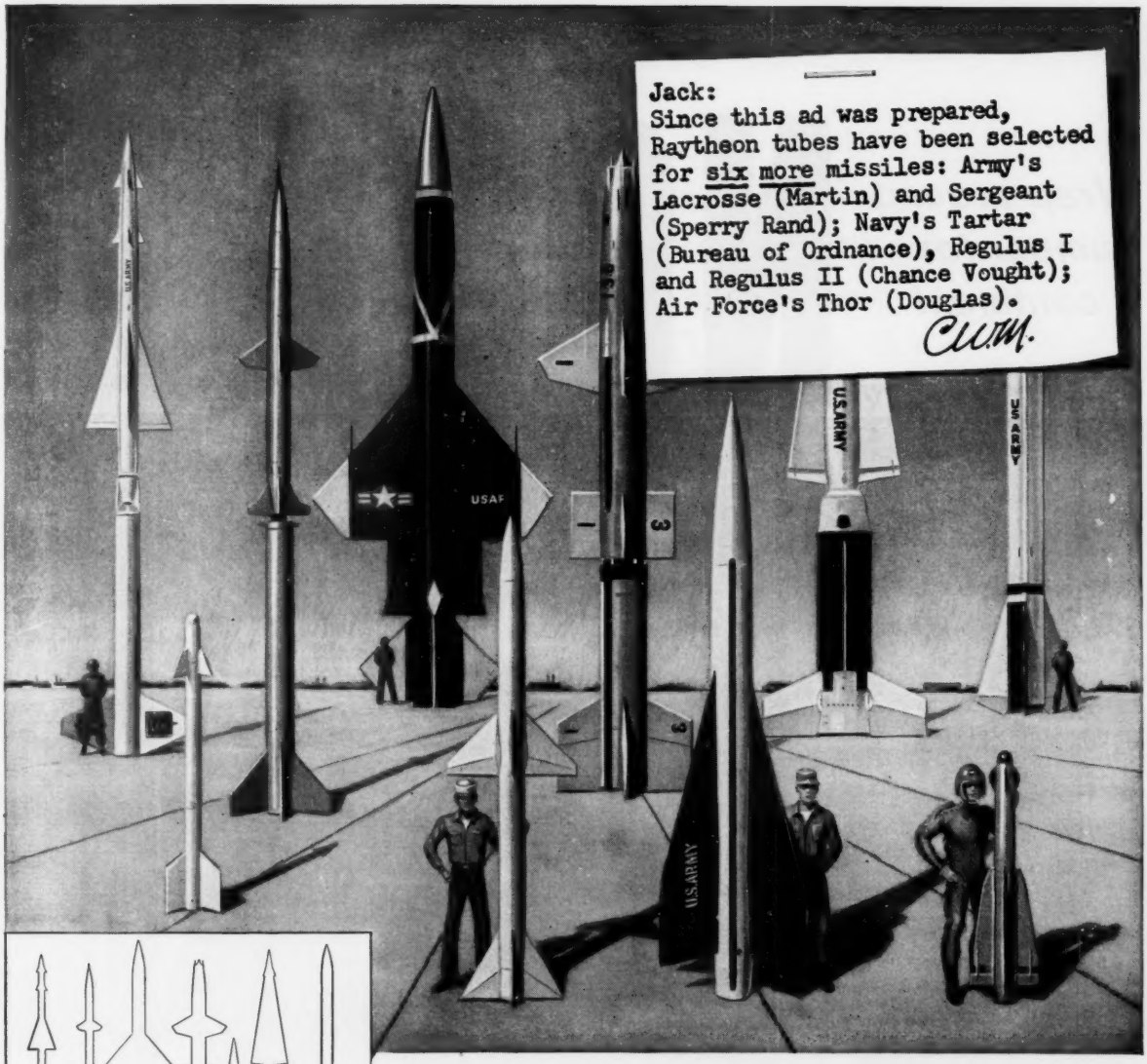
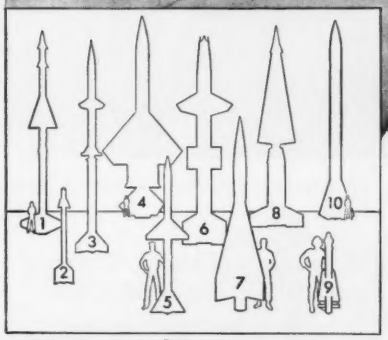


Jack:
Since this ad was prepared, Raytheon tubes have been selected for six more missiles: Army's Lacrosse (Martin) and Sergeant (Sperry Rand); Navy's Tartar (Bureau of Ordnance), Regulus I and Regulus II (Chance Vought); Air Force's Thor (Douglas).
CWM.



1. **Nike Ajax.** Army. Ground-to-air. Prime contractor: Western Electric.
2. **Sidewinder.** Navy. Air-to-air. Prime contractors: Philco; General Electric.
3. **Terrier.** Navy. Surface-to-air. Prime contractor: Convair.
4. **Bomarc.** Air Force. Ground-to-air. Prime contractor: Boeing.
5. **Sparrow III.** Navy. Air-to-air. Prime contractor: Raytheon.
6. **Talos.** Navy. Surface-to-air. Prime contractor: Bendix.
7. **Hawk.** Army and Marine Corps. Ground-to-air. Prime contractor: Raytheon.
8. **Nike Hercules.** Army. Ground-to-air. Prime contractor: Western Electric.
9. **Falcon.** Air Force. Air-to-air. Prime contractor: Hughes Aircraft.
10. **Corporal.** Army. Ground-to-ground. Prime contractors: Firestone; Gilfillan.

10 MAJOR U. S. MISSILES RELY ON RAYTHEON TUBES

Crushing acceleration and searing heat must be endured by the electronic tubes in guided missiles. Even under these grueling conditions, tiny Raytheon tubes produce guidance impulses with steadfast *reliability*. This reliability is achieved through capable engineering and painstaking manufacturing and testing techniques.

The choice of Raytheon Reliable subminiature tubes for use in these 10 missiles is another example of how the 31,000 men and women of Raytheon are contributing to the nation's security.

RAYTHEON

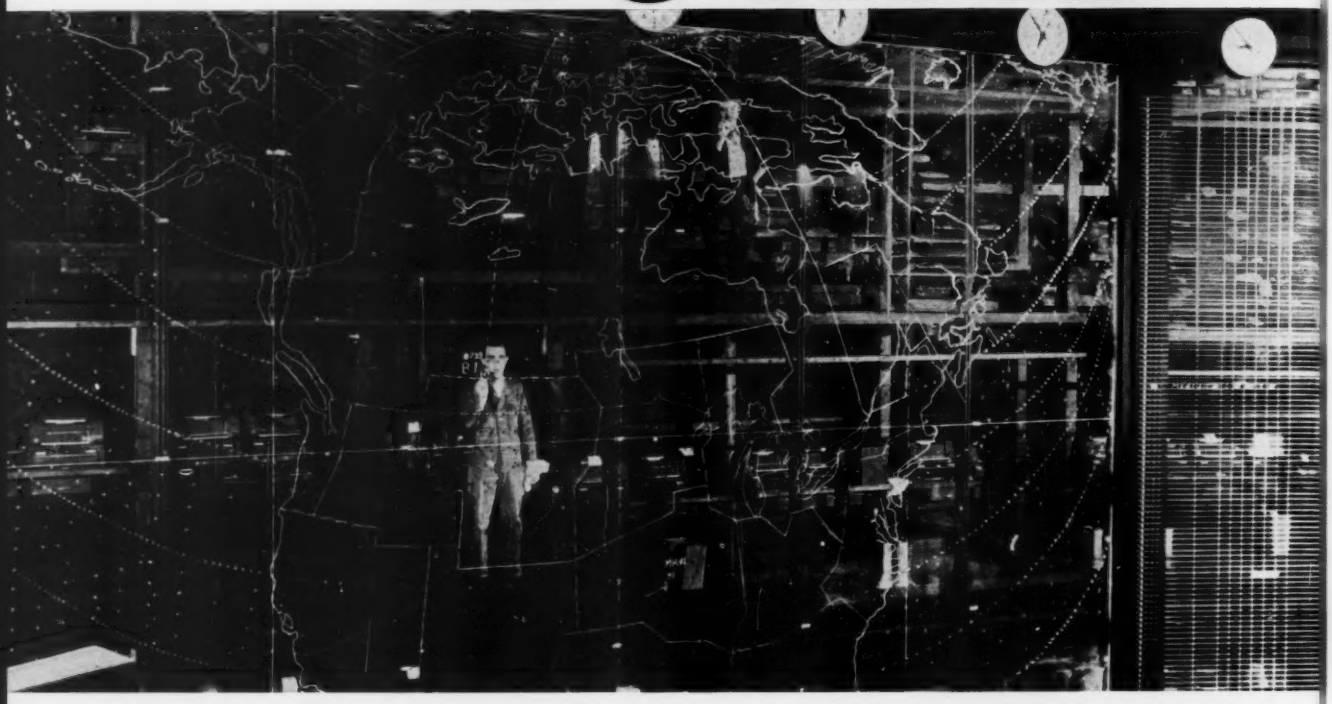
Excellence in Electronics

RAYTHEON MANUFACTURING COMPANY, Waltham, Mass.

Preserver of Peace . . .

Wrap-around bumper for a continent

NORAD



Headquarters—NORAD—Colorado Springs

Like a huge "bumper" wrapped around the North American continent and reaching down along both the Atlantic and Pacific shores, the North American Air Defense Command (NORAD) has been created for operational control of air defense units of the Army, Navy and Air Force of the U.S. and the RCAF Air Defense Command of Canada. Its field includes the vast area between the southern border of the United States and the northernmost limits of Canada and Alaska. Under the functional control of NORAD will be BMEWS (Ballistic Missile Early Warning System) and SAGE (Semi-Automatic Ground Control Environment) for the defense of specified sectors. In addition to its responsibility as prime contractor for BMEWS, the Radio Corporation of America is working on other important electronic assignments for NORAD.

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RADIO CORPORATION of AMERICA

DEFENSE ELECTRONIC PRODUCTS

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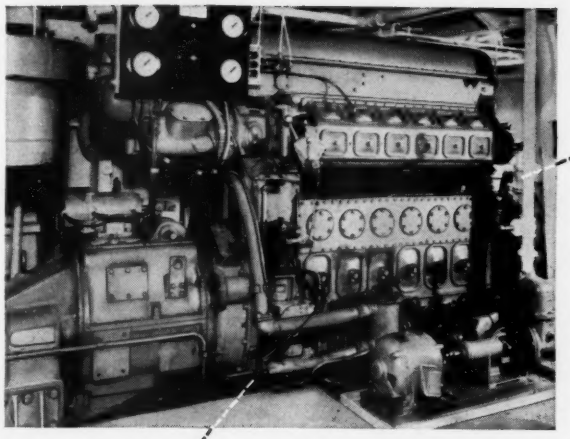

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—so says the president of Equitable Equipment Company who built this rugged, multipurpose craft in 1955.

The R. B. Kelly is serving as cargo ship, supply vessel, A-1 water carrier, tug boat, or operational barge serving offshore installations of the Pure Oil Company in the Gulf of Mexico.

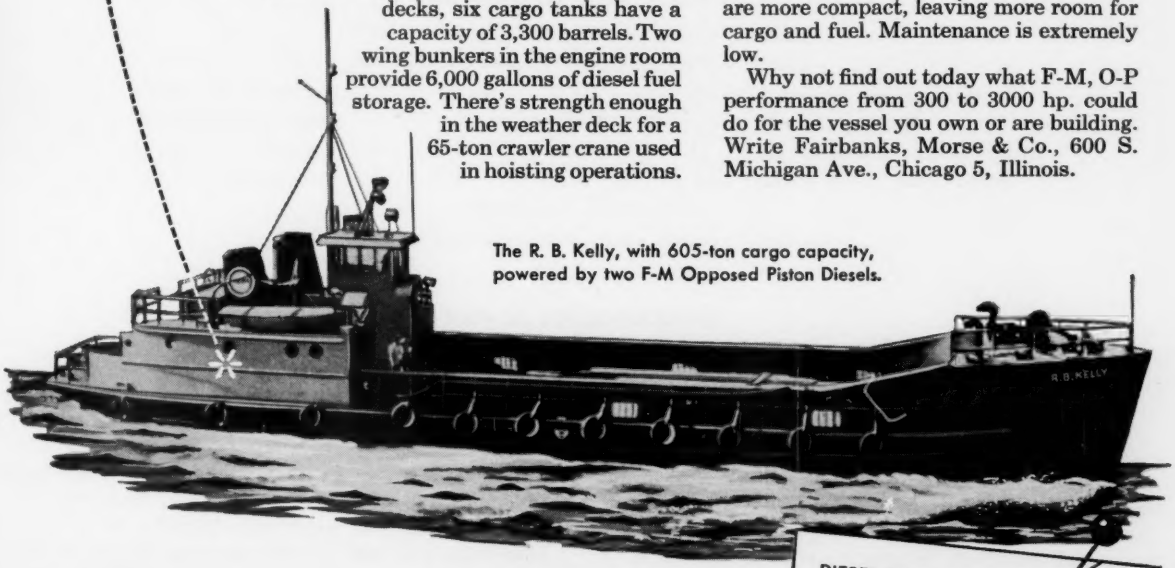
The Kelly can carry 10,000 feet of 9⅝" O.D. casing at a single loading. Below decks, six cargo tanks have a capacity of 3,300 barrels. Two wing bunkers in the engine room provide 6,000 gallons of diesel fuel storage. There's strength enough in the weather deck for a 65-ton crawler crane used in hoisting operations.

The R. B. Kelly is powered by two, 6-cylinder 38F5-¼ Fairbanks-Morse opposed piston diesel engines. Each one is rated at 450 hp. at 1200 rpm. Together the engines afford a line pull of 25,000 lbs. Her maximum speed is 12½ knots.

More and more vessels today in all waters are turning to F-M Opposed-Piston Diesels as the finest source of power and economical operation available. They are more compact, leaving more room for cargo and fuel. Maintenance is extremely low.

Why not find out today what F-M, O-P performance from 300 to 3000 hp. could do for the vessel you own or are building. Write Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Illinois.

The R. B. Kelly, with 605-ton cargo capacity, powered by two F-M Opposed Piston Diesels.



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Journal of the AMERICAN SOCIETY of NAVAL ENGINEERS, Inc.

VOLUME 70 NOVEMBER, 1958 NUMBER 4

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(Under whose supervision this number is published)

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NAVAL ENGINEERING

Definition: NAVAL ENGINEERING is the art and science applied in the design, construction, operation and maintenance of naval ships and their installed equipment.

SECRETARY'S NOTES

Meetings

For the first time, as a result of by-laws changes which were adopted last year, a meeting of the Council and the general Annual Meeting were held at the same time. It was, of course, necessary to develop a procedure for this and the method which was used seemed to be quite successful.

The Council meeting was held first with other members in attendance at the Annual Meeting as observers. These latter were given some opportunity to speak but voting was restricted to Council members.

After formal adjournment of the Council meeting, without changing positions or rearrangement of any kind, the President called the Annual Meeting to order. Notices of the Annual Meeting had been mailed to all members whose mailing address is in the Washington area. There are more than 400 of

these and rarely is the number who respond by attending the meeting more than ten percent. We suppose that this can be taken as an indication that most members are entirely satisfied with the way the Society is being administered and we derive some satisfaction from this. However, since the Annual Meeting usually conducts no business except to approve the slate for the annual election, it then becomes an open forum where any member present can, and frequently one does, express opinions or advance ideas which either start something or start a germination process which leads ultimately to an improvement in the Society or in an enhancement of its aims.

For example, at this year's meeting, Mr. Ralph Miller brought to our attention a growing clamor to establish some kind of an engineering or scientific award to be made in the name of the United States to someone who is identified with a noteworthy accomplishment. Mr. Miller pointed out that the pressure for such an award seemed to be coming largely from those identified with political interests and he feared that the award might be established and then administered as a political toy.

It is not easy to describe the concern which Mr. Miller expressed but it was easy enough for those who heard him to sense that the matter might become important. He urged that the Society join with other technical groups to act in such a way as to insure that if political forces did establish a United States Scientific and Engineering Award, it be administered in an absolutely sound and objective manner so that anyone honored by it and his accomplishment would be truly representative of the high professional standards which motivate true engineers and real scientists.

This, of course, is a very broad subject and one which the Society could not immediately act on. The concept is one which is well outside of the conventional operations of the Society. It could not even be crystallized enough to put on the agenda for future consideration. However, it was placed in the minds of those at the meeting and as an idea for the Society to consider, will either gestate in some of these minds until a definite course of action can be discerned or it will die. In either case, a thought was passed on to a few auditors and because the Secretary-Treasurer was one of these, it is here being recorded and passed on to all readers of the JOURNAL.

Definition of "Naval Engineering"

As previously reported, the Council had directed that the President appoint a committee to consider a definition of Naval Engineering, and if possible to recommend a definition which could become Society-official in time to be used for selecting a candidate for the 1958 ASNE award.

The committee which the President appointed was composed of

Rear Admiral W. A. Dolan, Jr., USN
Mr. J. J. Kennedy and
Captain E. A. Wright, U.S.N.

The Committee submitted a definition which was discussed widely by the Council and some of the other members present participated in the discussion. Slight modifications were made and a definition was formulated and approved by the Council with the limitation that it was to be placed before the membership at the Annual Meeting who would determine its adoption. This definition was accepted and approved by the membership who made up the Annual Meeting and who, under the By-Laws have full authority to act for the Society. Thus the Society now has an official definition of "Naval Engineering" which is printed just ahead of the Secretary's Notes.

During the discussion it was pointed out that this definition is much broader than the aims of the Society as described in Section 2 of the By-Laws. It was also pointed out that this apparent inconsistency was no inconsistency at all since the Society was free, if it chose, to define the recipients of an award which it made in any way which it decided upon. However, it seems reasonable to expect that, as this definition stands, if it is accepted without objection by the Society, the By-Laws will be amended at some future date to bring them into agreement with the definition.

ASNE Banquet 1959

The Council with not much discussion and by unanimous vote decided that:

a. There shall be an ASNE Banquet in 1959.

b. It shall be held on Friday, 1 May 1959.

c. It shall be held at the Statler Hilton Hotel in Washington, D.C.

d. Non-member guests will be limited to one for each two members except that the Banquet Committee may administratively rule on the fractional application of this to very small parties.

e. The cost shall be the same for members and for non-member guests.

f. Attendance shall be limited to the capacity of the facilities which the hotel can make available.

g. All other matters shall be left to decision and arrangement by a banquet committee to be appointed by the President.

The banquet committee will soon be appointed. We will probably mail applications for the banquet toward the end of February 1959. Until then there will be no arrangements for handling reservations and any which may be received before the regular forms become available with the announcement will be disregarded.

1958 Elections

At the Annual Meeting the report of the nominating committee was received. This committee had

been appointed by the President to nominate members as candidates to fill positions which become vacant by expiration of term on 31 December 1958. The Committee was composed of Vice Admiral Ralph E. McShane, U.S.N., Retired, Chairman and Captain C. H. Campbell, USNR and Mr. Phillip N. Israel as members.

Committee nominations were:

For President for calendar year 1959
Rear Admiral William A. Dolan, Jr., USN.

For Secretary-Treasurer for calendar year 1959
Captain J. E. Hamilton, USN, Retired.
Captain Robert B. Madden, USN

For Regular Navy Officer Members of the Council for 1959 and 1960
(Two vacancies)
Captain Thomas W. Rogers, USN
Captain Edgar H. Batcheller, USN
Captain James A. Brown, USN
Commander John A. Coil, USN

For Naval Reserve Officer Member of the Council for 1959 and 1960
Commander Robert S. Burpo, USNR
Commander Ralph S. Lorimer, USNR

For Civilian Member of the Council for 1959 and 1960
Mr. Gilbert S. Frankel
Mr. Don Caffery Glassie

There will be no vacancy of a Coast Guard Member during 1959.

The Committee's report was accepted and no nominations were offered from the floor before the nominations were closed. The ballot which has been mailed to all voting members and many of which have already been returned, carries the above names. Final counting of ballots will take place on 27 December 1958, reported to the Council about 1 January 1959 and will be announced to the membership in the JOURNAL of February 1959.

State of the Society

The Secretary-Treasurer made a brief general report to the Annual Meeting as regards membership and finances.

The membership survived the initial effect of the increase in dues (from $7.50 to $10.00 on 1 January 1958). The loss due to resignations at the beginning of the year, when 1958 bills were received, was a little bit heavier than normal. However, it was offset by the addition of new members and by 1 April 1958 had passed the first of the year count.

At the present time there are well over 200 members who are delinquent in their 1958 dues. This is larger than our normal condition and may probably be due in part to the increase in dues. Since, under the 1957 changes in By-Laws, members are dropped after one year's delinquency instead of two as formerly, there may well be a considerable drop in membership count as of 31 December 1958. This is quite unpredictable. Each day the mail brings three or four payments of dues for 1958 and it may well be that when we tot up the final account, we will find that losses have not been very heavy.

The membership has increased by an average of 58 per year over the last 50 years. Most members seem to be satisfied with this modest rate. Were all members in agreement with Captain Robert M. Barnes, U.S.N., Retired, we would have enjoyed a ten-fold increase in 1958 alone. Captain Barnes, in his normal day to day contacts presented the merits of the Society to more than ten eligibles whose applications were soon received. With a potential of many hundreds of thousands to work on, this same action is readily available to any member who believes in the Society enough to sell it to his friends and associates.

As to finances, the Society is sound if not very affluent. In 1958 we are operating under a formal, Council approved budget. Barring some completely unforeseen influence, the year's operations will show a slight gain. We will have spent less than our income. This will reverse the trend of the last few years when the Council advisedly ate into our reserves, with the former $7.50 dues, in order to establish an improved JOURNAL and to take care of the increases in administrative costs.

Although the year's operations will be at a slight gain, the book value of the Society will show a loss for 1958 because we are going to clear the books of all questionable accounts receivable. Some of these have accumulated during recent years and continuing to carry them presents a false picture of the Society's real worth. The loss will be a paper one and will indicate no decrease in true assets.

Dropping of Members

Some members do not seem to understand the effect of our By-Laws on the termination of membership. It is not possible to resign one's membership unless dues accounts are paid up to date. We have a very lenient policy of accepting resignations during a year if the resigning member has paid the pro-rata share of the year's dues. The only method of termination with an unpaid account, excepting death, of course, is to be dropped from the rolls for delinquency to which a certain amount of stigma is attached. Until this year members' accounts were permitted to run for two years. This has been changed so that only one year's arrears of dues is permitted.

To give members every opportunity to remain in good standing each one whose dues are unpaid receives two notices after the initial one and on 1 October, in addition, receives a final notice by registered or certified mail. It is, naturally, our desire to retain all members and the longevity of many of them indicates that the Society does very well in this regard. It is also very distasteful to add

"dropped for non-payment of dues" to any account, yet as we have described this becomes unavoidable in many cases.

Although there is some stigma attached to being dropped, the Society attaches no permanency to this. It can be removed at any time by the payment of arrears either with or without reinstatement of membership in good standing.

Addresses of Members

It is very disheartening to see the heavy return mail, both of JOURNALS and correspondence, on account of incorrect addresses. We know of no way to avoid this without the full cooperation of individual members.

The most valuable possessions of the Society are its bank account and money assets and its membership and mailing lists. The money accounting offers no problems. Methods of keeping this straight with the help of our bank are conventional, easily auditable and easy to manage. Not so with the membership list. For a few active duty Naval and Coast Guard officer members, if mail is returned, with some trouble and expense it is possible in Washington to obtain correct addresses. This probably fits less than one-fourth of the mailing list. We have no way to correct wrong addresses except for the member to tell us about it. Sometimes we can get word to the member by sending him a first class letter to his last known address. Some of these are forwarded whereas JOURNALS are not. However, even some first class and registered mail is returned. When the Post Office system admits failure in this way, perhaps we should accept this as evidence of the member's desire to be dropped but this we do not like to do because there is always the possibility that this is not the intent.

We again plead for each member, personally, to keep us informed of every change of mailing address. Only by 100 per cent compliance in this regard can the Society maintain the integrity of its membership list.

Administrative Assistant

On 8 October 1958 the Society entered its third era. The first lasted 20 years and might be named the Foundation era. The second which endured for 50 years is beyond any question the Fessenden era and the one just beginning, for the present must be called the post-Fessenden era. It will probably earn another name as it matures.

The breaking point between the first two eras was, of course, the hiring of Arthur Fessenden as Clerk. The second transition point was the retirement of this same Arthur Fessenden as Administrative Assistant after 50 years.

We knew that the day would come when the Society would have to continue without Mr. Fessenden and about a year ago, we employed a bookkeeper to learn the ways of the Society while the master was on hand to show her the ropes. We have strong evidence that she was an apt pupil and that the business affairs of the Society are in good hands since Miss Ruth M. Leonard assumed the title of Administrative Assistant on 8 October 1958. She with her recently acquired general Assistant Mrs. Ellis will constitute the office staff and will be successors to Mr. Fessenden. To what extent Miss Leonard becomes a full *replacement* only she and time can tell.

The Fessenden Tribute Dinner

Following the Secretary's Notes is a short factual biography of Arthur G. Fessenden. In that is allusion to a party which was given in his honor on 2 October 1958. Certain things about this party did not fit into the biography. Since they deserve recording we will mention them here.

The Council of the Society sponsored the dinner and underwrote the cost in the name of the Society.

The principal agent of the Council was the Statler Hilton Hotel where everyone, from the General Manager, Mr. Herbert C. Blunck down, went all out to make it the finest party which could be arranged. The general arrangement by the Sales Manager Wally Fauquier, the menu by the Maitre 'de Mike Tristano, the food preparation by the Chef Emile Buisson, the hotel facilities by the resident manager Jim Duchscherer, and the detailed arrangement of the South American room by the Second Floor Manager Walter Kotchin, have never been equalled in the memory of any one who participated. To top this off the bill which was rendered gave clear evidence that on the support side the Statler was very definitely a host.

All of the printing, invitations, mailing and that sort of thing was done for us by the Bethesda Business Service, whose owner is a second generation Fessenden. Here again the bill was of such a size that George Fessenden must be listed as a material host.

Certain of the groups in Washington which through the banquet and general society business have learned to depend heavily on Arthur Fessenden and have come to respect him very deeply, took some of the financial burden off the Society by sizeable contributions. These were the representatives of some of the larger industrial organizations in the country:

Buffalo-Forge Co.—Buffalo Pump Co.
Cla-Val Co.
The General Electric Company
Ingersoll Rand Co.
The Radio Corporation of America
Western Gear Corp.
The Westinghouse Corporation
Worthington Corp.

The result of what was really a spontaneous outpouring to honor Mr. Fessenden is that the total cost to the Society was less than we netted from the two sales of old JOURNALS which were among the last business deals which Arthur engineered before his retirement.

The guest list included seven past Presidents and two past Secretary's. Admiral J. D. Richardson, who was the senior past Secretary in point of Society service to attend, made the presentation of a silver bowl which was the memento given to Arthur as a token of the Society's love and respect.

Both of the Fessenden sons, George and Robert and one grandson helped their father and grandfather to enjoy a *Fessenden standard dinner*—perfect as to food, as to arrangements, as to good fellowship, as to spirit.

Manuscripts

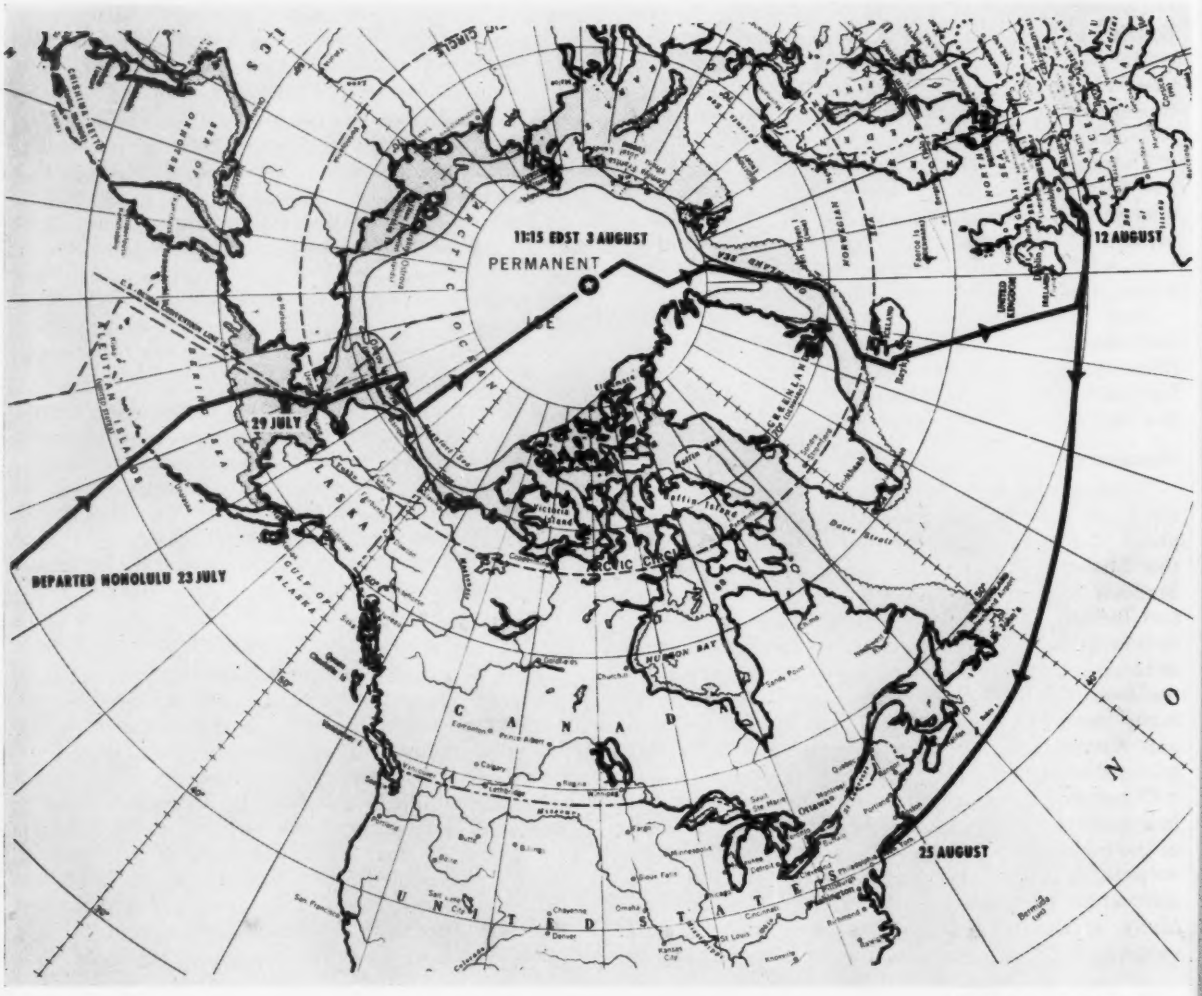
Readers of the JOURNAL are aware that in each issue we publish about twenty articles and that about half of these are original and copyrighted by the Society. The reprints are selected by Capt. Madden from among a wide selection of domestic and foreign periodicals and present to our readers in a form for permanent retention, the best of what is being printed in the English language throughout the world. Selecting the 10 or so *best* from among many hundreds is not an easy task but we feel that our Assistant Editor does a remarkably effective job in this regard.

Occasionally we ask someone who is best-versed in a subject to write an article for the JOURNAL but in the main our dependence is on those authors who voluntarily submit their manuscripts for our consideration. In treating these we admit some bias. Many ideas and experiences have very limited value in the world unless someone records them in writing. A very few will get an immediate response. The majority of them become permanently recorded for the benefit of future researchers and their value in this regard is not measureable.

The bias which we mentioned is in favor of *first* writings by Naval Members or those who have the qualifications for Naval membership. We feel that one of the purposes of the JOURNAL is to provide an outlet for these authors as an encouragement to them to continue their development as technical writers and to improve the facility with which they make their records. Some thoughts are passed along by word of mouth but this coverage is transitory and very narrow. The written word reaches a much wider field and it is permanent.

At the moment the volume in incoming manuscripts is very gratifying. We own a comfortable back-log of these so we do not encourage authors to help us for the sole purpose of filling the JOURNAL as was the case not so long ago. The great disadvantage in our present condition is that the timeliness of many articles is important and it is not easy to get the timely papers out on time when the medium is printed only quarterly. We are perforce limited to our own capacity to accomplish this.

It is hoped that the authors whose manuscripts we have bought will read this paragraph and will understand why they do not find their names in the Table of Contents. This is because we have not yet found room for them. We make every effort to get an article into print not later than the second JOURNAL issue after acceptance but we are not always successful in this. All articles which we have on hand and accepted now will certainly be published in the next two issues. Those on hand and not yet acted on, may, if they are accepted, have to be held for several months before we can print them.



Official U. S. Navy Photograph

THE ARCTIC CRUISE OF THE *NAUTILUS*

This plot of the route taken by the USS *Nautilus* on the transpolar cruise which disclosed a new "North East" passage is not unknown to most of our readers. It has been published in many periodicals and journals yet to none does it better belong than in this JOURNAL. We believe that the American Society of Naval Engineers should enshrine it in its permanent archives, this JOURNAL.

The plot epitomizes two things: the pioneering spirit of men of action and the advancing spirit of men of ideas. Columbus, as a man of action in 1492, could not visualize that some day this chart would be a reality. Perhaps Jules Verne, a man of ideas, in 1869 could have.

The bridge between the idea and the chart is spanned by the field of Naval Engineering. Nineteen years after Jules Verne's *Nautilus* was disclosed to the world, the American Society of Naval Engineers was founded in a world of coal-fired, fire-tube boilers and horizontal reciprocating engines. A mere 70 years later the successors of those 15 men, whether members of the Society or not, have brought forth as hardware the Rickover *Nautilus*. Many members of the profession under the direction of Vice Admiral (then Captain) Hyman G. Rickover, U.S.N. contributed. Many of them will not identify themselves with the Naval Engineering Profession. Yet the Society must claim them as certainly eligible as members.

What Admiral Rickover has spearheaded in 1958 is the counterpart of the coal-burning reciprocator of 1888. It is difficult to visualize how the growth curve of technology can continue at the phenomenal rate of the last 70 years. It would seem that it must be approaching an asymptote. Only Naval Engineers of the next 3 or 4 generations can determine what the naval world will look upon in 2028.

THE ASNE AWARD

We have discussed the ASNE award for well over a year. It has been reported and is well known to some that we went through the motion of making the award for 1957 but this was abandoned before finalization when the Council decided that insufficient groundwork had been laid in advance.

An award *will be made* for 1958 unless no nomination supporting adequate merit is received. This can happen, we believe, only if the members are indifferent. Certainly in the year 1958, American citizens have made substantial and remarkable contributions in the field of Naval Engineering whose definition will hereafter ride at the JOURNAL'S masthead and is repeated here:

NAVAL ENGINEERING is the art and science applied in the design, construction, operation and maintainance of naval ships and their installed equipment.

The award will go to the American citizen who made the most outstanding contribution in the field of Naval Engineering during 1958 **and whose accomplishment is known to the Awards Committee and to the Council.** We have underlined the critical element.

We are writing letters to a large number of officers and officials whose official positions put them where they can observe large segments of the field and we feel that few of them can have failed to note one performance which he considers to be outstanding. We are asking each of these to report this performance whether it be an idea, an invention, a concept, a writing or a physical deed.

By these letters we expect to cover a wide field and we hope that the response will reflect a full appreciation of what the Society intends its award to be.

But that kind of coverage cannot be complete. We are calling on 3400 members of the Society, to each of whom this notice is personally addressed, to report in the form of a citation any truly outstanding contribution which has come to his attention.

We know that many momentous things are being done. We hope that the most valuable of these will be cited to the Council so that the Award Committee will be very hard pressed to make the one outstanding selection.

Please address your citations to the Secretary-Treasurer (ASNE Award)
American Society of Naval Engineers
Suite 1004
1012 14th St., N.W.
Washington 5, D.C.

so that it reaches us not later than 15 March 1959. The citation must be thorough enough so that the Committee can make a proper judgement on it.

The success of the ASNE Award will depend to a very great degree on the response by the Membership to this appeal.

ARTHUR G. FESSENDEN

ARTHUR G. FESSENDEN

This article was prepared by the Secretary-Treasurer, whose eighteen years of close association with Arthur G. Fessenden probably exceed those of any other member of the Society. It was written on behalf of the entire membership. Although the Secretary takes full responsibility for the interpretations which are a large part of this biography, full credit for phraseology, diction and the other things which lend to its readibilty is due to the Assisttant Secretary-Treasurer, Captain R. B. Madden, and to Mrs. Pearl Vittucci of the Bureau of Ships, who read and revised the original manuscript.
It is our feeling that the Membership would want us to record in the pages of our JOURNAL, *something about this man Fessenden who has been so much a part of the Society for 50 years. We have done what we could to perform this very pleasant task for the Society and offer it as a permanent record.*

50 Years in Contemplation

UPON COMPLETING its seventieth year, the American Society of Naval Engineers enters the third era of its existence. This is marked by the retirement, on the eighth of October 1958, of Arthur G. Fessenden. It would be difficult to depict the life of Arthur G. Fessenden independently of that of the Society. It would be impossible to write the history of the Society independently of Arthur G. Fessenden. These two lives became closely merged and for the space of 50 years, one-half a century, they were partially interdependent.

This interdependence was not on equal terms. The Society gave to Arthur a portion of his income and it gave him an opportunity to become acquainted with, and known to, a group of men associated with the Naval Engineering profession. His contribution to the Society is not so easy to state in words. In taking advantage of his opportunity and merely by being himself he made for himself a host of friends and admirers. In so doing, he took the Soul of the Society, nurtured and protected it and assured forever, unless his precepts and examples are discarded, an honorable place for the Society among the many groups of men who are banded together for a purpose.

To those who believe that God will protect the right and that He chooses strange ways to do this, His hand can easily be seen in the manner whereby the Society and this man came and remained together. It is not a tale of glamour and of physically heroic deeds. In fact, unless one looks for true bases, it would appear to be quite prosaic. It is the purpose of these words to state the historical and biographical facts and to leave it to the reader to provide the analysis which best fits his own character.

MOLDING OF CHARACTER AND CAREER

The juncture of the two "A's", AGF and ASNE, occurred in 1908. Until that year they led separate existences, for practically all of which neither knew the other. Because of this it is only natural that the two should be described separately. Each should be understood in its own ambiency. It also seems natural that the older of the two should be recounted first.

If the life of Arthur G. Fessenden has been written by others it is unknown and unavailable to us. No searching inquiries have been made of old records; no unearthing of old friends and relatives for anecdotes has been made. What follows is almost entirely based on observations and casual conversations over a period of eighteen years and the distillation, from this collection of rather minor, in themselves, things, of the conclusions of the observer. In the writer's opinion they are factual. They are set forth as such for the reader's consideration.

Arthur makes much of the fact that he is a Yankee when, in fact, he is one only by adoption. He was born on the seventh of August 1879 in the City of New York—on Manhattan. In spite of adopted views and attitudes, he has retained the heritage of character which is often associated with the Metropolis; the knowledge of what is nice among the material things of life, and the cosmopolitan attitude derived from knowing and appreciating many types of people.

Indescribable is the effect when Arthur casually mentions either his mother or his father. His remarks do provide the matrix of a picture which helps to explain much of what Arthur himself is. As to the background of his parents we have not inquired.

His father was a career railroad man—a locomotive engineer. Shortly after Arthur's birth, his father transferred from the Central of New Jersey to the New York, New Haven and Hartford and moved his family from Manhattan to New Haven, which gave rise to Arthur's claim to Yankeeness. Arthur spent many hours in the cab of his father's locomotive, probably enthralled by the grandeur of it and eavesdropping on the conversation of the engine and train crews,-unconsciously absorbing their qualities of devotion to duty, their matter-of-factness in overcoming obstacles, and their honesty. He drank in these exemplary qualities and became imbued with those qualities which have adhered to the present day and will be with him to the end, —the qualities of LOYALTY, INTEGRITY and PERSEVERANCE. The traditions of railroading were noble ones and under their shadow Arthur was formed into the man we know.

The picture of his mother is not easy to set forth. We have heard no description of her from Arthur. When her name comes out casually in conversation it is without the bombast or the extravagant praise that one often encounters. It is in a tone and manner which, coupled with the Arthur whom we know, brings out a picture of a woman inherently honest, gentle, kind and with a deep-seated religion and love of humanity. Certainly from some source Arthur gained these qualities and it is easy to explain this on the basis of heredity and the environment of his childhood home.

Among the more mundane characteristics which Arthur most certainly picked up from his mother is his great love for ice cream. One gathers that seventy years ago Arthur and his mother "knocked-off" many a quart unassisted. Since he retains this great love for ice cream, and since this is quite easily and positively traceable to his mother, it seems reasonable that many of the other more important traits of character that he has should be ascribed to this same benign influence.

Passing hurriedly over the next few years we come to the point in his life where Arthur must look to the future and plan his career. He had developed into a baseball player of some ability and certainly considered becoming a professional ball player. However, his more conservative side led him to a more stable and socially more exalted profession—the bar. He entered Yale University and earned his LL.B. and he was admitted to the bar in the State of Connecticut in 1901.

Now we come to one of those inexplicable quirks which enter so many lives and turn them into most unexpected and certainly unpredictable channels. Arthur fell in love,—the object, one Lucy Wallace.

The economic position of a young lawyer in the early years of this century was not too secure. If successful he climbed to great heights but this took time. Arthur did not have time. He was in love.

Opportunity beckoned in the form of a Federal Government position. This paid a regular salary and in such a position Arthur saw the chance to lay up a nest egg which would allow him to marry without awaiting professional success.

The young lawyer in 1905 taught himself typing and shorthand and successfully passed the examination which qualified him for a government position as clerk, qualified as stenographer and "typewriter." The Civil Service Commission certified him to the Navy Department whose Bureau of Steam Engineering required a stenographer and "type-

THE FESSENDENS, Robert, George, ARTHUR, Roger

writer." On the 25th of January 1906, Arthur accepted and entered into a position in the Bureau of Steam Engineering at an annual salary of $840. He took this job for six months, to accumulate a little money to permit him to return to New Haven, get married and resume his chosen profession—the Law.

Now the Navy played a trick on him. On February 1st, just six days after he began, his salary was increased to $900, and 40 days later on 12 March 1906 it was raised to $1000. His plans had been toyed with, and finally they were completely shattered. With such wealth and such glimmering prospects, his resolutions evaporated. He changed his and Lucy Wallace's plans, swept her into marriage on June 4, 1906 and transported her to Washington, which became their "temporary" home for 28 years.

Just as an occasional remark has outlined a picture of his mother, so has a picture of his wife developed. The man was fortunate in his women and as we hear him mention them in casual conversation, each of these two must have had a feeling of pride at knowing the love and respect which each enjoyed from a great man.

Before she was taken from him by death in 1934, Lucy and Arthur enjoyed a very normal, perhaps somewhat routine, but ever idyllic love and marriage. Two sons were born to them, George on 11 April 1907, and Robert on 12 July 1910. Each of these, still living in Washington, bears tribute to his father in the inherited attributes of honesty, integrity, perserverance, diligence and loyalty. To my mind George Fessenden's business motto, "The House that Service Built" epitomizes his father, whose main objective in life seems to have been to *serve* in any position which he occupied.

Each of Arthur's sons has married and like his father has had two children,—a total of three boys and one girl. The first of these has recently married and Arthur will very probably soon see the appearance of another Fessenden generation.

One who has been fortunate enough to see the three generations of Fessendens, to note the feeling of love and respect which exists among them, is inclined to dwell on this patent evidence of the true worth of Arthur Fessenden. But we have much ground to cover and must continue this sketch of a man by merely mentioning this evidence of his fine character.

CIVIL SERVICE CAREER

When Arthur Fessenden came to Washington for a six months' "visit" in 1906, he came to the Bureau of Steam Engineering in the Navy Department. He joined an organization composed of himself and only 50 other people. By amalgamation and reorganization this Bureau became the Bureau of Engineering when it absorbed some of the functions of the Bureau of Equipment, which was abolished in 1913; and, in 1940 when the Bureau of Engineering and Bureau of Construction and Repair were joined, the

combination became, as it is today, the Bureau of Ships.

In 1906 the total population of these three Bureaus was 181. When Arthur retired after 39 years service in 1945, the Bureau of Ships had a population well up in the thousands.

The make-up of the 51 people who constituted the Bureau of Steam Engineering in 1908 was:

12 Naval Officers
5 Civilian officials who were those with an annual salary of $2000 or more
12 Clerks (into which category Arthur fell)
15 Professionals who were principally draftsmen, and
6 Laborers which included the messengers.

It is apparent that this must have been a very select group.

We obtained the official personnel record of Arthur G. Fessenden in the hope that it would prove to be a mine of information. Such is not the case. It contains practically nothing but a series of appointments and of periodic efficiency ratings. It shows nothing of the intimacy that developed between Arthur and Rear Admiral R. C. Griffin, who was the World War I Chief of Bureau. This intimacy grew up as a result of a man with heavy responsibilities finding a man of great integrity, with a legal mind and training and a great gift for assembling details into conclusive dissertations. This situation existed through the years with many others in the Bureau, in addition to Admiral Griffin. Yet it was never officially recognized by according Arthur the position which he truly held. Nor was this entirely the fault of the Bureau and its people. One who knows Arthur has no difficulty in realizing that his innate modesty and true humility kept him back from, in any way, pushing his own cause. He seems to have had full contentment from the knowledge that the position which he held gave him a job which was well done. His satisfaction came from the assurance that anything he had to do was done perfectly and, of much greater importance, that he had the respect and affection of everyone with whom he dealt.

His official record shows only two events out of the routine. In 1911 he was commended by the Chief of the Bureau for having, entirely on his own, and much of it outside of regular working hours, assembled the information from all over the world and set up, for the first time in the Bureau, a complete record of all motor boat engines and a system for perpetually maintaining this record correct. His methods were effective and used for 35 years. To our knowledge his was the first central inventory control system ever installed by the Navy. The important thing about it is that it is an example of a man seeing the need for something and pushing through against indifference and lack of support until he, by his sole efforts, had effected that thing. This is a typical "Arthurism."

The other event was the recommendation after World War II that he be given the Meritorious Civilian Service Award. The importance of his work to the war effort and the quiet effectiveness with which it was performed were recognized by his supervisors in the Bureau. This recognition should have given him deep satisfaction but because of its lack of glamour and the quietness and self-effacement with which the job was done, the award was not granted.

After his early meteoric rise from $840 to $1000 a year in a matter of a month and a half, his Civil Service Career was unmarked by any extraordinary advances. To those who are familiar with the pre-World War II situation when the clerical and administrative service was separate from the professional, and, in the Navy Department, the entire CAF series was limited at the top by the classification of one man and from there on down a hierarchical chain, his terminal grade of CAF-8 can be understood. Judged in the light of today, his ability and service and duties would have resulted in a much more exalted classification. His efficiency ratings through the years are ample evidence of the thoroughness and effectiveness of the way he did his job and of the esteem with which his seniors regarded him. He never put forth any claim to recognition for his educational and professional qualifications. As one evidence of his proficiency, this lawyer-turned-clerk was promoted during World War I into a technical position. His duties did not change but his status did because his supervisor recognized the technical nature of the clerical job which he performed. After the war, as jobs were reestablished on a permanent basis, his classification reverted to the clerical where it remained until the completion of his 39 years of civil service.

Arthur always worked in the Bureau of Steam Engineering or its successor Bureaus but for a period, forgotten by him, when the Secretary of the Navy was abolishing the Bureau of Equipment by administrative action in advance of legislative abolition. During this period Arthur was paid by the Bureau of Equipment and carried on its rolls as "assigned" to the Bureau of Steam Engineering. This was in 1913.

We will return to his retirement from civil service later. In the meantime, having laid the scene and having tried to show what manner of man he is in his personality and methods of doing a job, we must consider his connection with the American Society of Naval Engineers. To do this well it is necessary to set another scene—the status of the Society in 1908.

CAREER IN THE SOCIETY

In the Fall of 1908 the ASNE was twenty years of age. From a beginning in 1888 by 15 Naval Engineers, it had grown to a membership of 602 and had a net worth of about $9000. All of its affairs were

directed by a Council and executed by one man—the Secretary-Treasurer. He had no assistance, paid for by the Society, except the printer who printed the JOURNAL. Beside his own efforts he was entirely dependent on the facilities of the Bureau. His official office was the Society office. The Society records were kept by him in a pocket notebook. The membership list was maintained by the printer who mailed the JOURNAL. His stenographic assistance was provided, on the side, by the Bureau personnel who handled his official work. The Society's stock of back JOURNALS and reprints were stored in space which was provided by the Bureau. The scene was pretty well set for complete collapse. All that was needed was to have a Secretary-Treasurer who could not carry the quite heavy load of responsibility which rested on him alone. This condition arose in 1908.

The Council, recognizing that the job, with the expansion of the Society, had gotten beyond the capacity of a single person unless he was an extremely strong and able one, took action which was long overdue and decided to employ clerical assistance on a part-time basis. In early 1908 a civilian clerk in the Bureau was hired for $25 a month to assist the Secretary-Treasurer. Their first choice was a good and able man who resigned after a brief time. The job was offered to and accepted by Arthur G. Fessenden. He undertook this task on 8 October 1908. A few days later the Council accepted the resignation of the Secretary-Treasurer and appointed Lieutenant Henry C. Dinger to the position.

The new team of Dinger and Fessenden found themselves with a list of members of the Society and of subscribers to the JOURNAL and of a number of dollars. The records did not match the two. Who had paid for what and who had not was something of a puzzle, whose solution was Arthur's first job as Clerk to the Society. At the same time that he was doing his inventory job in the Bureau he worked on a very similar job for the Society, and by corresponding with each member and subscriber he brought the accounts into order. In 1912 he proposed a card record system which the Council approved and this method of keeping Society records has prevailed to the present. In the 46 years of his stewardship, Arthur's accounts have never been in doubt.

The Society lived through two World Wars under the nominal care of its Council and Officers but really with Arthur Fessenden's guidance. This has been attested by those Secretary-Treasurers who had the benefit of intimate association with him. The opinion of these officers is well described in the following letter which was received from Admiral S. M. Robinson, USN (Ret.). Admiral Robinson served the Society as Secretary-Treasurer from 1921 to 1923 and as President in 1931 and 1939.

"I have just mailed my card stating that I cannot attend the dinner to honor Mr. Fessenden but I cannot let this occasion go by without acknowledging my debt to Mr. Fessenden. When I was ordered to Washington to head the Design Division, I was practically ordered by the late Admiral Hepburn to take over the job of Secretary of the Society; neither I nor the Chief of the Bureau was keen to have me take this job as it was thought I would be too busy with my regular job. However, I took it and not only had no regrets but much pleasure and added to my professional knowledge. The reason for this was the unfailing assistance and advice of Mr. Fessenden.

"I never knew him to make a mistake and his advice was always very sound and I listened to it very carefully. I suspect that previous and succeeding secretaries did the same thing and that no doubt has had a great deal to do with the peaceful and progressive growth of the Society and the JOURNAL.

We all owe a great debt of gratitude to Mr. Fessenden and here's hoping that he may have many years to enjoy his well earned retirement.

Very sincerely,

S. M. ROBINSON."

Arthur has his faults, of course. The greatest and most annoying of these which I encountered was that when we differed, as we did occasionally, regardless of how the decision went, ultimately he was proven to have been right. No exception to this comes to my mind.

Once Arthur made the remark that he had some regret that he had not followed the legal profession but that this was not a very deep regret. In analyzing his relationship to the Society, I came to the conclusion that he had been very fortunate in the way his life had turned out. This cannot be discussed without some praise of the Society itself. The best attestation to the dignity and integrity of the American Society of Naval Engineers and of the great majority of its individual members is that Arthur Fessenden stuck to it for a half a century.

In any job or profession into which he entered, he could never have been happy if it meant the slightest compromising of what his conscience told him was RIGHT. In the Society he found the same kind of ideals which he himself held and followed. There was no object of personal gain; no material advancement at the expense of someone or something else; no backbiting; no jockeying for position; no walking close to the edge of the law for greater advantage; no acceptance of something base for a chimerical and hypothetical gain which in itself might be desirable. Here was a group of fine men, banded together for a single purpose which was not sought by politics or "politicking" or by any of the questionable practices which seem to be all too common either among men within a group or for the advantage of the group as a whole.

Arthur had no difficulty with his conscience in going all out, in giving everything he had to this group. It turned out to be an ideal and happy com-

bination. He lived with it for half a century, protecting it as it grew slowly but steadily from a membership of 602 to more than 3400; a net worth from $9,000 to $50,000. He saw its business grow from the requirement for a staff of two part-time employees to three full-time and two part-time. Never was it permitted to deviate from the ideals upon which it was founded and much of this was due to his behind-the-scenes influence.

A second and perhaps most glaring fault is his modesty which approaches humility. He has truly guided and not led the Society. He will not permit himself to be put out in front and he can always find someone to give the credit to for any of his ideas. He may not fully realize how many Secretary-Treasurer and Council decisions are the ones he would have made had he permitted himself to be put in a position of final authority—but others will.

DEVOTION TO THE SOCIETY

When World War II ended Arthur faced a problem. Only he knew what a drain on him the Society job was. The small personal type organization which he had joined in 1908, the Bureau, had grown into a partially impersonal giant. His Bureau job demanded his full time but less of him. It was apparent that he could not continue both jobs without serious detriment to one or both and he could not let himself be identified with a job done less than with human perfection.

To the Society's great benefit and gratification and, I believe credit, Arthur chose to continue with the Society and to take voluntary retirement from Civil Service. The decision was reached at some immediate monetary sacrifice and with a considerable decrease in his future retirement income prospects. I doubt if Arthur really evaluated this sacrifice because his love for the Society had become a very real thing and this and the associations which he continued undoubtedly meant something to him not measurable in dollars.

Thus for a few months more than thirteen years, Arthur devoted practically his entire life to the affairs of the Society. Although he retired after 50 years of employment by the Society a very conservative estimate based on consideration of vacations not taken and a conventional 40-hour week and average absences for sickness, would place his service to the Society as equivalent to about 70 years.

Although his life was one of devotion to his job he had limited outside interests. Perhaps the most important of these was his membership in the Masonic Order which he began in 1903. This interest in his life can be epitomized by the mere statement that in 1951 after holding many positions of honor and trust he was elevated to the 33rd Degree, the highest possible. This recognition of his character supports any statement which has been made here as to his quality.

RETIREMENT AND REPLACEMENT

Something of the man can be seen in his response when he was invited to attend a dinner by the Council to commemorate his 50 years of service. He accepted but with the statement, "Honors undeserved are degrading." Although this statement is correct its inapplicability was pretty well demonstrated by the response to the invitation to attend this dinner. This was extended to past officers and members of the Council and Banquet Committees who had known him most intimately in association on Society business. More than half of those who were invited from all over the world accepted and the 64 who joined him at dinner on October 2, at the Statler Hilton Hotel in Washington, attested conclusively to their belief, and the belief of the Society which they represented, that the concensus held Arthur Fessenden well deserving of any tribute or honor which might be accorded to him.

At this dinner, Admiral Leroy V. Honsinger, Society President, presided. He sketched Arthur's 50 years of Society life, not in terms of Arthur and the Society but of the world as a whole. He traced the many developments of the half cenutry from 1908 to 1958 and so established a very effective picture background on which to overlay the ASNE-FESSENDEN partnership and to fix very firmly in his listeners' minds just how long fifty years truly is. He first pointed out how far away the year 2008 seems to us to be and yet it is but fifty years into the future.

He then very effectively depicted how long ago fifty years in the past is by naming certain key events which had happened during Mr. Fessenden's years of service, and which to most of his hearers were things that their fathers or even grandfathers had told them about. The high spots of this calendar were:

In Arthur's first year, 1908, the Great White Fleet was half way around the world, having started the year before.

Also, in 1908, the Pearl Harbor Naval Shipyard, our first mid-ocean fleet supporting activity was established.

In 1909 Bleriot flew the English Channel on a pioneering flight and Peary reached the North Pole.

Many of the audience had been boy scouts yet this organization was not founded until 1910.

In 1911 an airplane first transited the United States from New York to Los Angeles in about four months elapsed time and 82 hours in the air.

The Republic of Free China was established in 1912, the same year which saw Scott reach the South Pole.

A year later this country gained its first income tax law and in 1914 the first ship transited the Panama Canal.

The first transcontinental telephone conversation was held in 1915.

Each of these events, when compared to conditions of today in similar fields, left little doubt in the minds of those present just what 50 years meant. This proved a most effective way to make people realize what is meant in human existence by a span of 50 years and to appreciate more fully exactly what Arthur had accomplished for the Society.

REPLACEMENT OUT OF THE QUESTION

The Society in 1956 unanimously elected Arthur an Honorary Member. His protest against this on the ground that he was not an engineer was swamped by the heavy vote of the membership. This recognized that whether or not an engineer by education or profession, he had done more for the welfare of the American Society of Naval Engineers than any other single man.

Arthur raised the question of his replacement in 1951. He seemed to feel that his age was a bar to utility to the Society. The thought of his leaving gave rise to consternation in the Council and more particularly in the mind of the Secretary-Treasurer. Much had to be done to prepare the Society for life without Arthur. In seven years the Council has acted progressively to prepare the way for his retirement, hoping that the actual day would be long in arriving.

Replacement was out of the question. You don't replace the Arthur Fessendens. What you do is to make physical arrangements for the continued performance of those necessary actions which he has been doing and hope that the elements of loyalty and devotion to duty and to a cause will spring up in some successor. Just taking care of his routine duties was a real problem. These things had to be continued and as realization came as to all of the things which he had been doing, it really seemed hopeless. These tasks in 1951 comprised: typing, filing, bookkeeping, running the checking account, keeping track of investments, acting as advertising manager with all of the details of contracts and copy and proof and billing, maintaining the membership records and the JOURNAL mailing list, setting up meetings and recording the minutes, building and maintaining good will, arranging practically all details of the annual banquet and keeping track of the Society's calendar. We have tried to arrange for the continuation of all of the Fessenden functions and enter the post-Fessenden era with hope that mistakes will be few and not too damaging.

The sound basis on which Arthur established each of his functions facilitates this task of arranging for their continuation. We are hopeful that his superior work will be continued with a quality such that the Society will survive and will continue to flourish as it did under his guardianship.

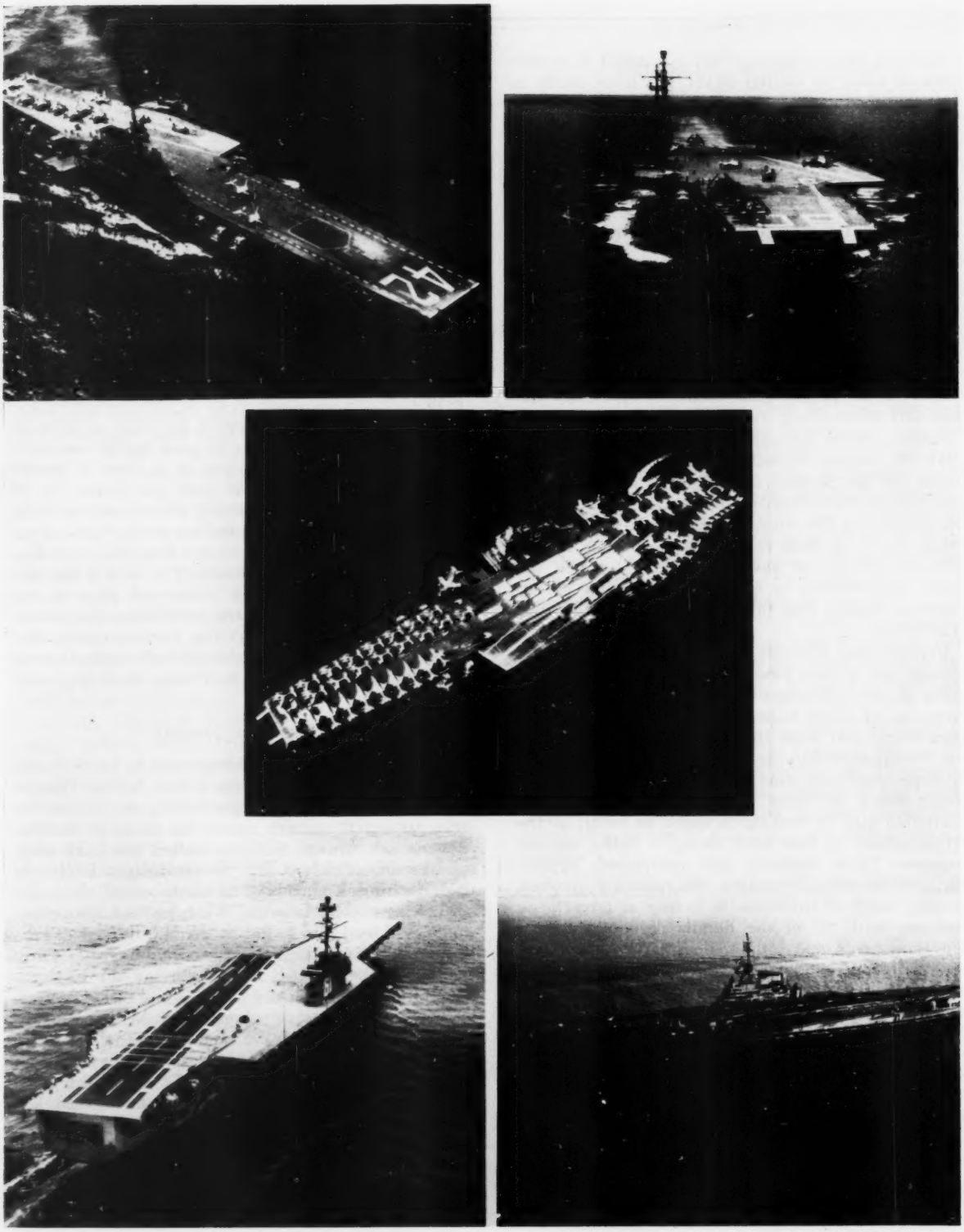
While this process was going on, with the specific hope by the Secretary-Treasurer that he personally would not hold the responsibility of that office when the day of separation actually arrived, Arthur frequently mentioned and suggested his retirement. This culminated in the announcement by him in early 1958 that his doctor had decreed that he must quit work no later than the following September. At that time he did seem to show signs that his health was not at its best. By mutual agreement the actual date of retirement was set for the 50th anniversary of his Society service which was 8 October 1958.

During the last few months any sign of physical failure disappeared and as he gave up the responsibilities of fifty years he was in a state of health which was the envy of at least one junior by 20 years. His physical agility and mental acuity made us wonder if we had not been too precipitate—if we should not have demanded more from him by keeping him on the job indefinitely. This selfish thought might have prevailed had things not gone to the point where transfer of responsibilities had to be done in justice to others. The Society grudgingly released Arthur so that he can devote some time to himself, for perhaps the first time since his early clerkhood.

A TIME FOR HIMSELF

Through the years we have come to know something about the simple things which Arthur Fessenden likes. He is very happy driving an automobile without much concern about the route or interim destination. Always duty has called him back after an absence of only a few hours. He now is free to indulge himself and to be as unconcerned about his return as about his route. We hope that his retirement will lead him to get farther away from Washington than his duties have permitted and that many years will be allowed him to pursue the exploration of the United States which is one of his desires.

The Society sincerely hopes that Arthur will enjoy many years of a full life and have the assurance that his newly gained freedom will be devoted, perhaps partially to a gratification of some of the pent-up desires which duty has kept from fulfillment, but largely in some way which he will find, to the betterment of his fellow men and the world in which they live.



Official U. S. Navy Photograph

Montage of aircraft carriers of the fleet shown are: Upper left, *USS Franklin D. Roosevelt* (CVA-42); upper right, *USS Saratoga* (CVA-60); Center, *USS Forrestal* (CVA-59); lower left, *USS Ranger* (CVA-61); lower right, *USS Midway* (CVA-41).

DAVID LINCOLN HERR

CONIC-SECTION ORBITS DERIVED FROM THE GRAVITATIONAL POTENTIAL

THE AUTHOR

Honorary and Naval Member of the Society, has been a commissioned reserve officer of the United States Navy for twenty years, five and one-half years of which were on active engineering duty in the World War II period, during which he received two Navy Letters of Commendation, serving under Vice Admiral E. W. Mills, U.S.N., Vice Admiral H. G. Bowen, U.S.N., and Rear Admiral H. G. Rickover, U.S.N. His seven years of academic preparation were at the Moore School of Electrical Engineering of the University of Pennsylvania, the Massachusetts Institute of Technology, Harvard University, and the Polytechnic Institute of Brooklyn, for which he holds three degrees obtained under seven National fellowships and scholarships. His specialties are design of military and scientific computers, inertial navigation, guidance and weapons systems, counter-weapons, precision computer components, transducers and physical sensors, and precision high-yield mass-production techniques and designs. Since receiving his baccalaureate and the A. Atwater Kent Award in Electrical Engineering in 1937, Mr. Herr's technical papers, systems and components designed, patents and patents pending number over fifty, including an advanced networks text with Dean Harold Pender. In 1945, he received the American Society of Naval Engineers Award for his Prize Essay on the subject "Engineering in the Navy as Seen by an Active Reserve Officer." In 1950, Mr. Herr was the recipient of the A. Cressy Morrison Award in Natural Sciences of the New York Academy of Sciences for his work on the theory, design and production of precision computer components. With Dr. D. B. Steinman, he was co-organizer and first secretary of the Academy's Section of Mathematics and Engineering. The New York Academy of Sciences designated Mr. Herr Organizing Chairman of the first international nuclear science congress to be projected here. Since 1953, after various professional and corporate associations, Mr. Herr has been consulting and designing engineer and president-owner of Mark Instrument Co., Inc., Santa Monica, California, engaged in the licensing of various of his new proprietary transducer and computer components. Since 1956, he has been airborne staff engineer, Librascope, Inc., General Precision Equipment Corp., Glendale, California. Most recently, in May, 1958, Simon Gratz Public High School, Philadelphia, Pennsylvania, gave him its Quarter-Century Distinguished Service Alumni Award, and in June, 1958, he served as Organizing Chairman and Moderator of the A.I.E.E. Summer General Meeting and Air Transportation Conference Space-Flight Control Symposium in Buffalo. Mr. Herr, a registered professional engineer, is a member of scientific, engineering, management and professional societies and committees, here and abroad, including the Royal Society of Arts (Great Britain), and of several industrial-military organizations.

DEDICATION

The Author writes spontaneously but most appropriately for this issue:

"I respectfully desire to publicly dedicate this latter paper to my fellow Honorary Member, Arthur G. Fessenden, who, this October 1958, completes 50 years of loyal, devoted, competent and never-failing service to our Society.

We, who know him for long, regard him respectfully and affectionately. I sincerely trust that he, and you, Sir, will accept, on behalf of the Society, this Space Age token of appreciation and esteem of Mr. Fessenden who, literally, has bridged the gap from Sail to Space in the engineering History of the United States Navy."

THE ORBITING mass, m, with velocity, v, is very small compared with the central mass, M, whose motion is assumed zero. No other masses are considered.

The gravitational potential energy and the velocity kinetic energy characterize the 2-body system entirely. The system is conservative, that is, there is no dissipation.

The kinetic energy is always given by $\frac{1}{2}mv^2$. The potential energy is always given by $-\gamma mM/r$, where r is the distance from the orbiting mass, m, to the central mass, M, in the point sense, and γ is the universal gravitational constant.

The total energy, U, is the algebraic sum of the kinetic and potential energies.

All unchanging, stable, conic-section orbits (circle, ellipse, parabola and hyperbola) are characterized by constant total energy, U.

The conic-section orbit is an hyperbola, a parabola, or an ellipse, according to the constant total energy, U, being greater than zero, equal to zero, or less than zero. The potential energy and kinetic energy relate to one another in such manner as to maintain the difference of their magnitudes, (K.E.+ P.E.), constant and greater than zero, equal to zero or less than zero, for the hyperbola, parabola and ellipse, respectively. In the circular orbit (a limit of the elliptic case), not only the total energy is constant, but so also are the kinetic energy and the potential energy. Further, for the circular orbit, the total energy is equal to ½ the potential energy and both are negative; the total energy and the kinetic energy are equal in magnitude and opposite in sign.

Correspondingly, the eccentricity, ε, of the hyperbola, parabola, ellipse and the circle, is, respectively, greater than unity, equal to unity, less than unity, and equal to zero.

The ratio of the kinetic energy to the potential energy for the hyperbola, parabola, ellipse, and the circle is, respectively, algebraically less than -1, equal to -1, greater than -1, and equal to $-\frac{1}{2}$. The parabolic and circular orbits are the only two conic-section orbits for which this ratio of energies, (K.E./P.E.), is constant, having the respective values of -1 and $-\frac{1}{2}$. In an elliptical orbit this ratio of energies oscillates between the maximum and minimum values of $-(1+\varepsilon)/2$ and $-(1-\varepsilon)/2$. In an hyperbolic orbit of eccentricity, ε, this ratio of energies varies between the maximum and mini-

TABLE A

Conic-Section Orbits Derived from the Gravitational Potential

CONIC	K.E.	P.E.	$U_{constant}$	ε	$-\left(\frac{K.E.}{P.E.}\right)$	$-\left(\frac{K.E.}{P.E.}\right)^{max}_{min}$	v^2	$v^2\Big]^{max}_{min}$
HYPERBOLA	$\frac{1}{2}mv^2$	$-\frac{(\gamma mM)}{r}$	$+\frac{(\gamma mM)}{2a}$ >0	>1	>1	∞ $\frac{(3+\varepsilon^2)}{4}$	$\gamma M\left(\frac{2}{r}+\frac{1}{a}\right)$	$\frac{\gamma M}{a}\left(\frac{\varepsilon+1}{\varepsilon-1}\right)$ $\frac{\gamma M}{a}$
PARABOLA	$\frac{1}{2}mv^2$	$-\frac{(\gamma mM)}{r}$	0	1	1, constant	1, constant	$\gamma M\left(\frac{2}{r}\right)$	$\gamma M\left(\frac{2}{n}\right)$ 0
ELLIPSE	$-\frac{1}{2}mv^2$	$-\frac{(\gamma mM)}{r}$	$-\frac{(\gamma mM)}{2a}$ <0	<1	<1	$\frac{(1+\varepsilon)}{2}$ $\frac{(1-\varepsilon)}{2}$	$\gamma M\left[\frac{2}{r}-\frac{1}{a}\right]$	$\frac{\gamma M}{a}\left(\frac{1+\varepsilon}{1-\varepsilon}\right)$ $\frac{\gamma M}{a}\left(\frac{1-\varepsilon}{1+\varepsilon}\right)$
CIRCLE	$\frac{1}{2}mv^2$, constant	$-\frac{(\gamma mM)}{R}$ constant	$-\frac{(\gamma mM)}{2R}$ $=\frac{P.E.}{2}$	0	$\frac{1}{2}$, constant	$\frac{1}{2}$, constant	$\frac{\gamma M}{R}$, constant	$\frac{\gamma M}{R}$, constant

mum values of $-\infty$ and $-(3+\varepsilon^2)/4$.

Finally, the quantity, v^2, the square of the velocity of the orbiting mass, m, is equal to the product of a constant, γM, by the factor $\left(\frac{2}{r} \pm \frac{1}{a}\right)$ In these expressions, "a" is ½ the distance between the orbital vertices. The plus sign is chosen for the hyperbola, the minus sign for the ellipse. For the parabola, "a" goes to infinity. For the circle, "a," the semi-major axis of the ellipse, goes to R, the circle radius.

The quantity, v^2, the square of the orbital velocity, has the following maximum and minimum values:

$$\frac{\gamma M}{a} \cdot \frac{(\varepsilon+1)}{(\varepsilon-1)} \text{ and } \frac{\gamma M}{a}, \text{ Hyperbola;}$$

$$\gamma M \cdot \frac{(2)}{(n)} \text{ and zero, Parabola,}$$

where "n" is the distance from parabolic vertex to focus;

$$\frac{\gamma M}{a} \cdot \frac{(1+\varepsilon)}{(1-\varepsilon)} \text{ and } \frac{\gamma M}{a} \cdot \frac{(1-\varepsilon)}{(1+\varepsilon)}, \text{ Ellipse;}$$

and, the constant value, without maximum or minimum, $\frac{\gamma M}{R}$, Circle.

For the ellipse, it is to be noted that the geometric mean of the maximum and minimum values of v^2 is simply $\gamma M/a$.

While the varying ratio of the kinetic to the potential energy of an elliptical orbit symmetrically transits the value of $-(\frac{1}{2})$, only the circular orbit is characterized by the *constant* value of $-(\frac{1}{2})$ for this ratio.

Table A summarizes these characteristics.

Table B summarizes the relative total energies, $U_{rel.}$, for all satellites known to have been successfully placed in orbit around Earth to 15 July, 1958, from VANGUARD I (taken as unity total-energy relative value) through EXPLORERS I and III to SPUTNIK III. Such relative total energies of different satellites to Earth are proportional to m, the mass of the elliptically orbiting satellite, and inversely proportional to the two-thirds (⅔) power of the time-period of orbit, T.

TABLE B

Total Energy of Elliptically Orbiting Earth Satellites

Earth Satellite	"m" (pounds)	T (minutes)	$U_{rel.}$, Relative Total Energy
V_I	3.25	134.0	1
E_I	30.80	114.5	10.524
E_{III}	31.00	115.7	10.519
S_I	184	96.2	70.613
S_{II}	1,120	103.7	408.83
(S_{II}*)	(7,000)	(103.7)	(2,555.2)
S_{III}	2,919	106.0	1,050.1

S_{II} Payload Earth Weight.

S_{II}* Total Earth Weight, Payload Plus Container.

In a study of the optimum trajectory of a rocket, Robert R. Newman of the Applied Physics Laboratory, Johns Hopkins University, concludes that, in order to maximize the range, the rocket should not have low acceleration in the dense air of lower altitudes and then increase its acceleration as it reaches higher altitudes, as commonly thought. Instead, it should have a higher acceleration in dense air and should reach the end of burning at a comparatively low altitude. Also, contrary to popular belief, the rocket should not climb almost vertically until it reaches thin air before rotating, in order to minimize the time and length of path in the denser part of the atmosphere. Instead, the rotation to about 45 degrees should be accomplished in a relatively short time in order to achieve maximum range.

—from JOURNAL OF THE FRANKLIN INSTITUTE

September 1958

A summary of the problems faced in high speed planing surface craft has been set forth by the staff writers of SHIPBUILDING AND SHIPPING RECORD. Three principal problems are involved. The first is the suppression of aerodynamic lift on the hull of the craft at high speeds. Means by which this can be done unfortunately lead to increased aerodynamic drag. The second major problem is the hydrodynamic phenomenon of porpoising. Attempts to analyze the factors involved have led to the conclusion that the effect is a function of planing area, planing angle, speed and load. No satisfactory method has been developed to insure the suppression of this tendency by proper design, though some progress has been made. The third problem is the structural problem of withstanding the peak loads produced by the extreme accelerations encountered in high speed travel over water. Accelerations of up to 8.5q have been recorded, and it is believed that even this value may be considerably exceeded under some conditions.

—from SHIPBUILDING AND SHIPPING RECORD
September 18, 1958

A specialized electron tube shape has been developed by Westinghouse Electric Corporation. Known as the "match box" design, the new configuration is relatively flat, with its leads protruding on the four sides of the "box." The elimination of the tube stem and the welding of the leads to the mount structure in one plane are said to improve the reliability of the tube. Other advantages noted are high dimensional accuracy of envelope, increased resistance to severe vibration and shock, increased flexibility of location in the design of equipments, easier handling in mechanized assembly operations, reduced space required and reduced noise and circuit capacitance. Promising for commercial use, the tube is being used so far in military equipments.

—from WESTINGHOUSE ENGINEER
September 1958

An unusual tanker built for the Esso Petroleum Company in the Monfalcone shipyard of the Cantieri Riuniti dell' Adriatico has provision for carrying pressurized petroleum products cargo. The 49,000 ton tanker is 690 feet in length and has a beam of 90 feet. Steam turbine driven, she is expected to make a trial speed of 17 knots. The unusual feature is the arrangement of pressure tanks for carrying the liquified petroleum gas products. A total of 58 tanks are designed for internal gas pressures of 150 psi. Forty of the cylindrical tanks are mounted vertically in two longitudinal rows in the center tanks of the ship. These vessels, or bottles, are nearly 17 feet in diameter and 51 feet high. They are surrounded by cargo oil carried in the center tanks. The remaining 18 bottles are arranged in two horizontal rows in the wing tanks. They range from 11 feet to 13 feet in diameter and are 35 feet long.

—from SHIPBUILDING AND SHIPPING RECORD
October 2, 1958

W. G. CAMACK and G. LEPPERT

AN EVAPORATIVELY COOLED REACTOR

ACKNOWLEDGEMENT

We are publishing this paper as a reprint without remuneration to the Authors. The paper was written for and was presented to a technical session on Experimental Power Reactors and Advanced Concepts at the 1958 Nuclear Science and Engineering Conference in Chicago on March 17 to 21, 1958. It has been published in pamphlet form on behalf of the Engineers Joint Council by the American Institute of Chemical Engineers.

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A NON-BOILING, EVAPORATIVE COOLING SYSTEM

As AN outgrowth of studies made of the behavior of a heated pool of mercury, a system has been conceived whereby useful quantities of vapor can be generated directly from a reactor coolant without ebullition in the reactor core. From data reported by Bonilla, et al., (1) * it was found that when mercury was heated at low pressure in moderately deep pools, high rates of heat transfer were effected without ebullition. Since the mercury was static (in a pool with a free surface), high rates of evaporation were necessarily experienced at the free surface. Such a non-boiling† evaporative system could be employed in a reactor power cycle (see Figure 1) by submerging the fuel elements in a pool of coolant of sufficient depth that an appreciable increase of saturation temperature would occur from the evaporating surface to the vicinity of the fuel elements. Then the desired heat transfer rate could be attained without bringing the temperature of any point on the heated surfaces in the core above the saturation temperature for the pressure at that point in the core. The resulting temperature difference between the heated surface and saturated vapor at the pool top (the over-all ΔT) would be the saturation temperature difference corresponding to the pressure change due to static head and would be a function of the pressure of the vapor evolved (as well as being a function of the heat flux) since $(dT/dP)_{sat}$ varies with pressure.

Temperature measurements made by the Bonilla group indicated that the bulk liquid temperature (away from the heated and free surfaces) was nearly constant with the depth. Their findings agree with those reported by Jakob (2) (3) on work done by Heidrich. It can be deduced from Heidrich's work that the temperature profile in an evaporating pool of superheated water is similar to that shown in Figure 2.

*Numbers in parentheses refer to similarly numbered references in bibliography at end of paper.

†Throughout this paper, "boiling" is used synonymously with "ebullition," as contrasted with "evaporation," which is used to define vaporization from a free surface without bubble formation.

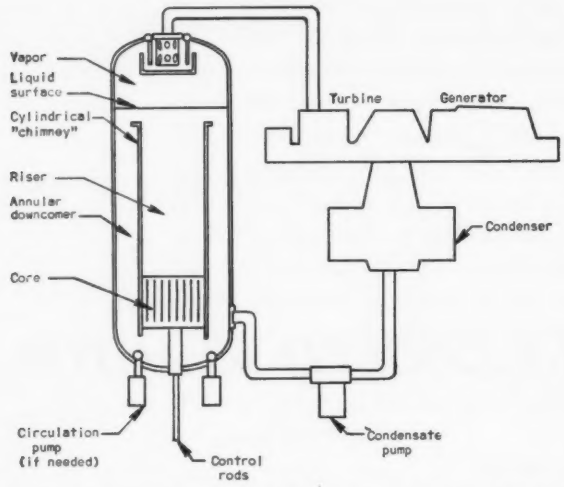

Figure 1. Evaporatively Cooled Reactor in a Power Cycle.

The temperature difference from the heated surface to the pool top is available to overcome the resistance at the two surfaces and to supply the temperature drop through the bulk of the liquid. As a consequence of the nearly constant bulk temperature (which is below saturation at the pool bottom and above saturation at the top), relatively large temperature differences are established between the bulk of the liquid and the heated and free surfaces.

The system under discussion would have the advantage over most reactor cooling methods that no secondary heat exchanger is needed. Pressurized water reactors, organic moderated reactors, sodium cooled reactors, etc., require a secondary heat exchanger to transfer the heat from the reactor coolant to the working fluid. On the other hand, in the proposed system, the vapor evolved could be utilized directly in a turbine.

Of course, the desirable feature of employing the coolant vapor directly in the turbine is to be had with a boiling reactor, too. However, because of the inherent variations experienced during boiling, stability problems arise in both thermal and fast reactors. In a thermal reactor, if the coolant acts as the moderator, as is the case in a boiling water reactor, vapor formation in the core effects moderator density variations which in turn vary the reactivity. Similarly, in a fast reactor, vapor formation leads to a variable reactivity because of the resulting fluctuations in poison density.

Even though boiling water reactors have been shown to be stable, the rate of vapor formation is very much restricted in that the vapor-liquid ratio must be kept low. No such restriction would be placed on the non-boiling coolant cycle. Furthermore, because the core would contain liquid only, its volume could be considerably smaller than that for a boiling reactor. In short, the proposed non-boiling, evaporative cooling system would be free of many of the design difficulties encountered in boiling systems.

FACTORS INFLUENCING THE CHOICE OF COOLANTS

The prime requisite of the evaporatively cooled reactor (ECR) is a liquid to meet the unique requirements that will be imposed on it. The rate of change of saturation temperature with liquid depth, which is determined by the quantity $\rho(dT/dP)_{sat}$, must be large in order to avoid excessively high liquid columns. Then too, the desired $\rho(dT/dP)_{sat}$ must be had in a reasonable temperature range: in the upper limit, the temperature cannot be more than, say, 1000°F (or the maximum allowable fuel element temperature), nor can the temperature be so high as to yield excessive vapor pressures. In the lower limit, the pressure at a reasonably low condenser temperature should not be too low, nor should the required sink temperature be below

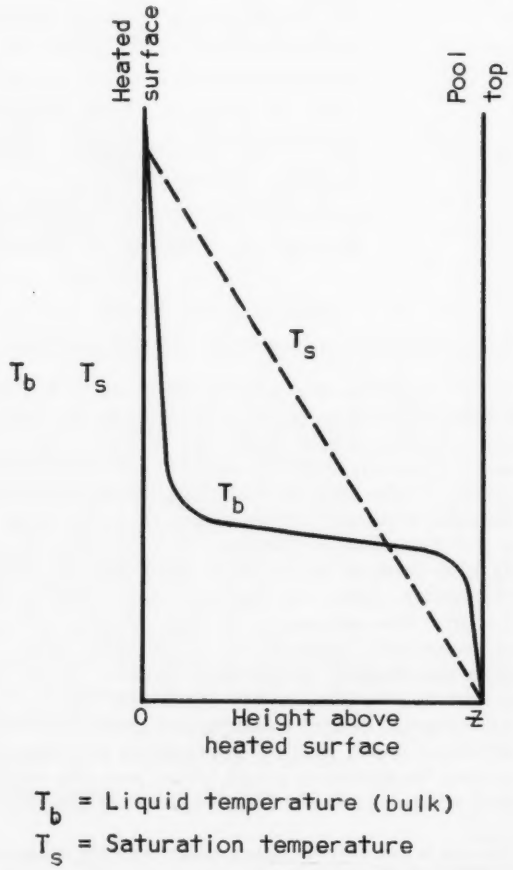

Figure 2. Temperature Distribution in Non-Boiling Evaporating Water.

what is commonly available. Further, the liquid must be stable under the conditions to which it will be subjected (e.g., it cannot decompose as a liquid nor can it disproportionate during evaporation).

The preceding specifications must be met regardless of the type of reactor. In addition, if the reactor were to be thermal, the coolant would have to have a low thermal neutron absorption cross section. It would also be fortunate if it were a good moderator as well. Then too, it would be desirable (for both thermal and fast reactors) if the liquid were readily available and cheap; if it were non-corrosive and non-toxic; if it had a low melting point and high latent heat of vaporization (for good thermal efficiency); and if its physical and thermodynamic properties were well enough known to facilitate preliminary design considerations.

EVIDENCE OF HIGH RATES OF HEAT TRANSFER IN NON-BOILING, EVAPORATING MERCURY

The possibility of obtaining appreciable evaporation rates without boiling at the heated surface was brought out during an analysis of a report by Bonilla, et al. They state in the report that the following may occur: "At low enough pressure and/or high enough depths—natural convection with no boiling at the heated surfaces, but with vaporization at the top surface...." They further state that "the mercury pressure in a boiling experiment was determined from the bulk temperature of the boiling mercury." Prompted by these statements, one can deduce that certain curves presented in the Bonilla report represent data for conditions of temperature and pressure under which boiling at the heated surface could not have taken place.

For example, the slope of their curve for a 10 cm deep pool at 4 mm pressure (see Figure 3) suggests that it may represent natural convection rather than boiling. This is shown to be the case in the following way: the pressure given, 4 mm, is the saturation pressure for the bulk temperature they measured, yielding a bulk temperature of 318°F. Now the maximum wall temperature reached in the experiment was the bulk temperature plus the maximum ΔT or 418°F. However, the pressure at the bottom of the pool was at least 100 mm (due to static head alone) which corresponds to a saturation temperature of 502°F. Therefore, since the heated surface never reached saturation temperature, boiling could not have taken place there. Nevertheless, the evaporation rates at the liquid surface were comparable to those obtained during boiling at higher pressures, even though the heater temperatures involved were not as high as might have been expected. For instance, a flux of 10^5 Btu/hr ft^2 was obtained with a wall temperature of 418°F at 4 mm pressure; the same flux is well within the nucleate boiling region at 1 atm with the same pool depth, as can be seen in Figure 3.

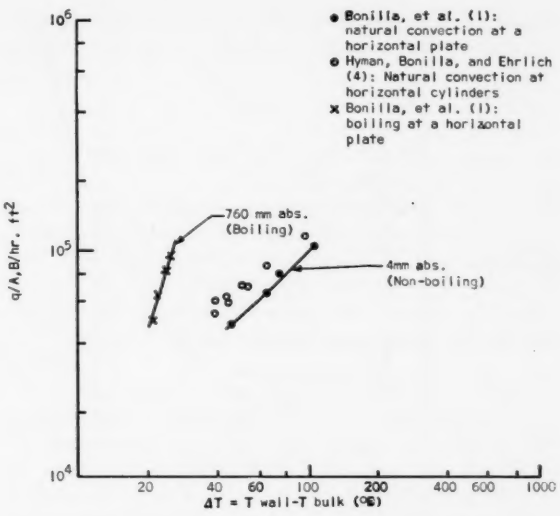

Figure 3. Heat Transfer to Mercury for Boiling and Non-boiling (Natural Convection) Conditions (1), (4).

Actually, all that is involved is normal natural convection in a pool pressurized by its own static head. In fact, the data given by the Bonilla group agree quite well with natural convection data for horizontal cylinders taken by Hyman, Bonilla, and Ehrlich (4). Some representative points from their work are also shown in Figure 3.

The basic principle involved, a self-pressurized pool, could be used in either a natural or forced convection system. However, for the time being, we will concern ourselves mainly with the natural convection process. An explanation of this phenomenon can be had by considering the basic relationship represented by the vapor pressure curve together with the information given by Jakob (2) (3) on Heidrich's work with heated pools of water.

INFLUENCE OF SYSTEM PRESSURE ON THE EFFECT OF STATIC HEAD

In any pool of liquid, the static head of fluid gives rise to a difference in saturation temperature between the top of the pool and the bottom. Depending on the pressure on the system, the difference may be large or small, as can be seen in Figure 4, which is a typical vapor pressure curve. For a system being operated at pressure P_1, with a pool depth such that the pressure differential is ΔP, the effected ΔT_1 is small. However, if the same pool is at pressure P_2, the resulting ΔT_2 is considerably larger. From this it follows that with high density liquids at low pressures, large differences in saturation temperature can be had with relatively small depths. However, large saturation temperature differences alone do not explain the high heat transfer

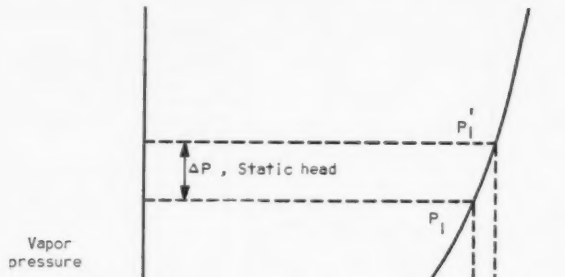

Figure 4. A Typical Vapor Pressure Curve.

rates; some knowledge of the actual temperature variation with depth is necessary for a full explanation.

OBSERVATIONS ON NON-BOILING EVAPORATION

Jakob (2) (3) gives an account of work done by Heidrich with non-boiling evaporating superheated water: Heidrich found that a temperature profile was set up in the pool of water as shown in Figure 5. The slope of the profile at the free surface depends strongly upon the heat flux, although the variation is not linear. In fact, Jakob, using Heidrich's data, deduced that the ΔT at the free surface was the same as that which would be had at a heated surface during boiling with the same heat flux.

In any event, it is easily shown that the temperature difference between the bulk liquid and the free surface (ΔT_1) is considerably smaller than the difference between the heated surface and the bulk liquid (ΔT_2). By using the equation given by Jakob for free convection in a pool,

$$\frac{hL}{k} = 0.16\left[\left(\frac{\rho^2 g L^3 \beta \Delta T_2}{\mu}\right)\left(\frac{\mu C_P}{K}\right)\right]^{1/4}$$

the following values of ΔT_2 were calculated for comparison with Heidrich's measured values of ΔT_1:

Heat flux	*Free surface*	*Heated Surface*
q″ (Btu/hr ft²)	ΔT_1 (°F)	ΔT_2 (°F)
1570	3.6	17
2000	5.1	20
5350	9.0	41

From the above observations it is seen that the main thermal resistance is through the lower liquid film. Consequently, the major portion of the over-all ΔT is at the heated surface. However, the over-all ΔT can be made to reach any practical value by varying the pressure on the pool and the pool depth; so the rate of heat transfer (and evaporation) is limited only by the allowable liquid column height.

Actually, there is a theoretical maximum evaporation rate which can be predicted by kinetic gas theory, but the rate is beyond anything commonly experienced. For example, water with an evaporating surface temperature of 212°F has a theoretical maximum rate of 7.1×10^4 lb/hr ft² as compared with the 1.62 to 5.53 lb/hr ft² given by Heidrich. The maximum rate would correspond to a heat flux of 6.9×10^7 Btu/hr ft.²

Before full use can be made of these findings in accurately predicting the behavior of the proposed non-boiling evaporative system, several points in question must be cleared up. For instance, since Heidrich's work was done with water, there might be some doubt as to how nearly his findings apply to other liquids, particularly liquid metals. A full evaluation of the non-boiling case would require experimental data on the entire temperature profile, which is not available from the accounts of Heidrich's work. Similarly, some means of predicting the ΔT obtained at the free surface would be needed. Furthermore, there is no assurance that behavior similar to that observed in small pools would be experienced in much larger vessels.

All of the uncertainties above would have to be resolved through research, as would the characteristics of the particular fluid chosen. Nevertheless, a

Figure 5. Temperature Distribution in Non-boiling, Evaporating Superheated Water (2).

preliminary evaluation of possible fluids can be made if certain assumptions, based on Heidrich's data, are made.

INVESTIGATIONS OF POSSIBLE COOLANTS

Since the necessary height of liquid is directly proportional to the slope of the vapor pressure curve at the operating point and inversely proportional to the liquid density, the initial search for the desired coolant was restricted to high density, low vapor pressure liquids. In order to stay within the specified range of temperatures and pressures, only liquids with vapor pressures lower than water but higher than one atmosphere at 1000° F were considered. The only elements that fall within this range are mercury, phosphorus, and sulfur; but several halides meet the requirements. Phosphorous was eliminated because of its erratic behavior of undergoing undesirable phase transitions. For example, at certain pressures it changes from liquid to solid and back to liquid in the region between 660°F and 1000°F. Likewise, sulfur appeared unpromising because of its viscosity, which can vary by three orders of magnitude between 250° and 425°F. For instance, in going from 320° to 355°F, the viscosity increases from about 55 lb/hr. ft to nearly 4×10^4 lb/hr ft.

Mercury is completely suitable as a fast reactor coolant for the system under discussion. Upon considering melting points, stability, and thermal neutron cross sections, the possible thermal reactor liquids were narrowed to several halides and diphenyl. Diphenyl was added for consideration, even though its density is low, because of its low cost. Other organics, notably the terphenyls, have been shown to be less susceptible to both pyrolysis and radiolysis than diphenyl. For example, Nakazato and Gercke (5) found no fouling during in-pile loop tests with a diphenyl and orthoterphenyl eutectic mixture. The exposure time was 11.2 days at the reactor power of 30 megawatts with in-pile heater surface temperatures of 450° to 700°F. The heat flux was 25,000 Btu/hr ft.^{2} Since thermodynamic properties were available and since its $\rho(dT/dP)_{sat}$ was the largest among the halides considered, bismuth tribromide was chosen as the most likely halide. Consequently, $BiBr_3$, mercury, and diphenyl were singled out for further analysis.

Of the various requirements imposed upon the coolant, that of a necessarily high $\rho(dT/dP)_{sat}$ is probably the most limiting. Both mercury and $BiBr_3$ are satisfactory at 1000°F, but diphenyl must be used at around 450°F to yield a reasonable $\rho(dT/dP)_{sat}$. Nevertheless, this criterion alone does not eliminate diphenyl, since its good moderating value at low cost renders it very desirable for a thermal reactor coolant. Mercury could be used only in a fast reactor, but $BiBr_3$ could probably be used regardless of the neutron energies since it has neither a high absorption cross section nor a high scattering cross section.

None of the three liquids chosen has a particularly high latent heat (cf. h_{fg}=970.3 Btu/lb for water at one atmosphere). The high melting point of $BiBr_3$ may make its use impractical, as may the scarcity of physical and thermodynamic data, but these difficulties alone may not disqualify it. However, the high present cost of $BiBr_3$ may prohibit its use. In any event, $BiBr_3$ shows some promise and has been included in the calculations of necessary column heights and thermodynamic (Rankine) cycles.

As a basis for comparison, an over-all ΔT of 50°F was chosen. For mercury in natural convection, Bonilla, et al., (1) found that a flux of about 5×10^4 resulted; for diphenyl and $BiBr_3$ the flux would be dependent upon the type of flow established in natural convection but would probably be reasonably high. However, it might be advantageous to supply forced convection to improve the heat transfer. In either case, the pool depth necessary to prevent boiling at a given pressure and for a given ΔT would be the same. As another basis for comparison, a 1000°F vapor temperature was chosen for mercury and $BiBr_3$, but since diphenyl is so light it was necessary to work with 425°F vapor to get a reasonable value of $\rho(dT/dP)_{sat}$. Table I gives the calculated minimum pool depths at the temperatures and pressures indicated. A sample calculation is given in the Appendix.

It will be recalled that the necessary head can be diminished greatly by operating at lower pressures (e.g., mercury at 30 psia and 750°F requires only a 2.75 ft pool for an over-all ΔT of 50°F). However, with diphenyl the pressure is already so low that it might be impractical, if not impossible, to expand the vapor through a turbine and derive any useful work. By using forced convection an acceptable heat flux could be had with a lower over-all ΔT, making it possible to operate at higher temperatures without increasing the necessary pool depth.

A simple Rankine cycle has been calculated for the same operating conditions used above. For purposes of comparison, an outlet turbine pressure of 0.2 psia has been chosen. With all three liquids, the sink temperature required is considerably higher than that which is commonly available; so a lower pressure could be used for higher thermal efficien-

TABLE I

Minimum Required Liquid Column Heights

Liquid	Operating Conditions			Necessary Head (ft)
	Pressure (psia)	Temperature °F	Over-all ΔT °F	
Hg	180	1000	50	12
$BiBr_3$	47	1000	50	12
Diphenyl	6	425	50	16

TABLE II—*Ideal Thermal Efficiencies for Hg, $BiBr_3$, and Diphenyl*

Liquid	Ideal Turbine Work $W_k=h_1-h_2$(B/lb)	Total Heat Input $Q=h_1-h_3$(B/lb)	Thermal Efficiency	Turbine Outlet Quality (x=lb vapor/lb mixture)
Hg	57	143	0.40	0.66
$BiBr_3$	18	34	0.53	0.60
Diphenyl	50	236.6	0.21	superheated

cies, if lower pressures could be tolerated. The exhaust pressure would probably be dictated by such considerations as vapor specific volume, quality, and the ability to seal the cycle effectively.

Rankine cycle values, along with ideal thermal efficiencies (for isentropic expansion) are given in Table II. The thermodynamic properties of $BiBr_3$ and diphenyl were extrapolated from existing data (6) (7) (8) (9).

It will be noted that diphenyl has the unusual and interesting property of being superheated upon expansion. This is an asset in that one of the limiting restrictions on the lower pressure for most substances is the turbine outlet quality.

The efficiency of the diphenyl cycle could be improved by adding a regenerator, since the superheated exhaust gas is hotter than the condensate. This would be especially true if the turbine inlet pressure were increased to a more reasonable value.

It is interesting to note that organics have been considered by others as possible thermodynamic fluids. Several years ago the Dow Chemical Company abandoned the idea of developing a Dowtherm turbine when difficulties arose in handling the gland seal at the turbine shaft (10). In addition, the availability of high pressure, high superheat steam has discouraged any further development on their part. More recently, Purdy (11) suggested the possibility of using diphenyl in nuclear power plants.

SUMMARY AND CONCLUSIONS

Studies of experimental work by Bonilla, et al., with mercury and by Heidrich with evaporating water led to the conception of a high rate, evaporative cooling method that could be used in a reactor system.

A non-boiling evaporatively cooled reactor appears to be feasible as well as desirable; it would have all the advantages but few of the serious disadvantages of a boiling system. The ECR would be particularly attractive if it were a fast, mercury cooled reactor, since mercury is the ideal fluid with respect to necessary pool depth and has good thermodynamic properties as well. Diphenyl is promising as a thermal reactor fluid, mainly because it is cheap and because it is a good moderator. Bismuth tribromide has a high thermal efficiency and low required pool depths, but it is expensive.

A fast reactor core submerged to a depth of 12 ft in mercury could produce 180 psia saturated vapor. Under these conditions, a saturation temperature difference of 50°F would exist in the pool. With natural convection this would permit a heat flux of about 5×10^4 Btu/hr ft.^{2}

A complete evaluation of the evaporatively cooled reactor will depend upon whether there would be a great advantage in avoiding vapor formation in a thermal reactor core if the liquid used were only a coolant, not a moderator. Likewise, it must be established whether there would be any disadvantage in boiling a fast reactor coolant in the core. The answer to both of these questions lies in an analysis of the effect of poisons and poison density variations on reactivity. No quantitative analysis has been made, but it seems from qualitative considerations that the net effect of possible density decreases (e.g., boiling) in the coolant would be an increase in reacivity with a possible subsequent "run-away" condition.

Some of the other basic unknowns in an ECR are the dynamics of natural convection in a large pool; the parameters affecting evaporation at a free surface; the temperature distribution in a large evaporating pool; and possibly the properties of the liquid chosen. For example, natural convection in a large pool may prove to be so disordered that it would be necessary to put the core in a "chimney" to provide a riser and down-comer, as Figure 1 indicates. Then too, it may turn out that natural circulation is not enough to pass the liquid through the core effectively. Both the "chimney" and forced convection could be used to the advantage of the system by assuring that only subcooled liquid would be circulated over the fuel elements.

In all the foregoing analyses, it was assumed that any desired evaporation rate could be effected at the free surface without any substantial change in the temperature distribution. However, this might not be the case. There is some evidence that the thermal resistance at the surface (and thus the ΔT) is a function of liquid purity (12) (13). Impurities tend to collect at the surface and might increase the thermal resistance in the free-surface film enough to cause the temperature difference between the bulk liquid and the vapor evolved to be a sizable part of the over-all temperature difference. This would mean that a deeper pool would be needed for the same heat flux.

NOMENCLATURE

h = coefficient of heat transfer, Btu/hr ft^2 °F
h (with subscript) = enthalpy, Btu/lb
h_{fg} = latent heat of vaporization, Btu/lb
P = pressure, psia, or mm of Hg, abs
ΔP = pressure difference, psia
q/A = q″ = heat flux, Btu/hr ft^2
s = entropy, Btu/lb °R
T = temperature: $T_{w,z}$ = temperature of water at depth z below surface

T_{st} = temperature of steam
T_b = bulk liquid temperature
T_w = heater temperature
T_s = saturation temperature

ΔT = temperature difference: $\Delta T_1 = T_b - T_{st}$, $\Delta T_2 = T_w - T_b$
$(dT/dP)_{sat}$ = change of saturation temperature with a change in pressure, °F/psia
x = steam quality, lb vapor/lb mixture
z = pool depth
ρ = density

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12. "Maximum Evaporation Coefficient of Water," by K. C. D. Hickman. Industrial and Engineering Chemistry, Volume 46, Number 7, 1954, pp. 1442-1446.
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APPENDIX

Sample Calculation of Minimum Pool Depth for an over-all ΔT at 50°F

For mercury at 1000°F, 180 psia (vapor evolved), the ΔP corresponding to a ΔT of 50°F is 70 psia (i.e., the saturation pressure at 1050°F is 250 psia).

The depth of mercury necessary to effect a pressure difference of 70 psia is:

$$Z = \frac{\Delta P}{\rho} = \frac{70\ \text{lb}}{\text{in}^2} \times \frac{144\ \text{in}^2}{\text{ft}^2} \times \frac{\text{ft}^3}{790\ \text{lb}} = 12\ \text{ft}$$

For mercury at 750°F, saturation pressure=30 psia; for mercury at 750°F ΔT=800°F, P_{sat}=45 psia

$$Z = \frac{\Delta P}{\rho} = \frac{15\ \text{lb}}{\text{in}^2} \times \frac{144\ \text{in}^2}{\text{ft}^2} \times \frac{\text{ft}^3}{795\ \text{lb}} = 2.75\ \text{ft}$$

A gas-suspension reactor coolant has been under investigation by engineers of The Babcock and Wilcox Company. Designed for high heat transfer properties at low pressures, a coolant consisting of fine graphite particles suspended in CO_2 has been the subject of detailed study. Mixtures with densities of up to 8 lb/ft^3, circulated at a pressure of 35 psi, temperature of 300°F and velocities of 40 ft/sec, have performed satisfactorily, giving a heat transfer rate of about eight times that of pure CO^2 under similar conditions.

—from BABCOCK AND WILCOX press release October 1958

A plastic material having a unique combination of physical properties has been recently introduced in this country. Developed in Germany and marketed in Europe, the substance, known as "Vulkollen," combines the properties of hardness, strength, rigidity, resistance to abrasion, oil and oxygen. It has been successfully used in motorcycle sprocket gears for operation periods of over 60,000 miles. Its good shock absorbent qualities make it specially suitable for use in handles for pneumatic tools and similar shock-mitigating applications.

—from JOURNAL OF THE FRANKLIN INSTITUTE
August, 1958

Believed to be the largest centrifuge in the world, an acceleration machine at the Naval Aviation Medical Acceleration Laboratory at Johnsville, Pennsylvania, consists of a 10 foot gondola mounted at the outer end of a 50 foot tubular steel arm. The gondola is completely instrumented and equipped with dummy flight control gear so that operations and reactions of a pilot can be observed in a high acceleration environment. Simulated space flight in the centrifuge has recently been achieved. The projected behavior of the North American X-15, designed to carry a pilot to the outer fringes of the Earth's atmosphere, has been programmed in the centrifuge in order to determine the mode of behavior of the pilot and to simulate flight conditions for pilot training.

—from ELECTRICAL ENGINEERING
October, 1958

A disposable, aluminized suit for protection of fire fighters has been designed and tested by the U. S. Army Engineer Research and Development Laboratories. The suits are made of flame retardant treated, aluminized kraft paper. A hooded parka, face mask, a pair of leg sleeves and a pair of mittens are provided for each suit. Initial tests indicate that men wearing the suits over duty uniforms were able to remain for two or three minutes without discomfort in a heat environment intense enough to blister exposed portions of helmet liners.

—Department of the Army press release
September, 1958

Before a joint meeting sponsored by A.S.M.E. and A.I.E.E., Mr. R. L. Schmidt of the General Electric Company presented a paper describing a merchant ship nuclear reactor installation study. Citing an annual increase for the ship of 29,000 tons cargo carrying capacity over that for a conventional ship of the same size and displacement, Mr. Schmidt estimated a possible fuel savings of 30 to 40 per cent lower than present costs. Chosen for the study of the 22,000 SHP propulsion plant was a boiling water reactor. Mr. Schmidt indicated that this selection was made because of its greater simplicity, high cycle efficiency, and present state of development.

—A.S.M.E. press release
September 30, 1958

RICHARD C. JOHNSON

THE COST OF FINISHES AND TOLERANCES

THE AUTHOR

has been in the office of Value Engineering since February 1955. Prior to this, he was employed for four years in the Special Projects Section of the Turbine and Gear Branch of the Bureau of Ships. Before coming to the Bureau, he was assistant to the Chief Engineer at Briggs Filtration Company, a field engineer for steam turbine installation with the General Electric Company and a design engineer for Reynolds Metals and Solvay Process Company. He completed a specialized course in Mechanical Engineering in 1941 at the University of Pittsburgh and is now completing a four year course in Business Administration-Management, at Southeastern University. During World War II he served as a fighter pilot in the U.S. Air Force.

INTRODUCTION

A GREAT DEAL of time and effort has been spent in cost reduction through the use of new materials, new processes, new production methods, and many other items, once the design has been established. But surprisingly little attention has been given to some of the factors in design which set these costs.

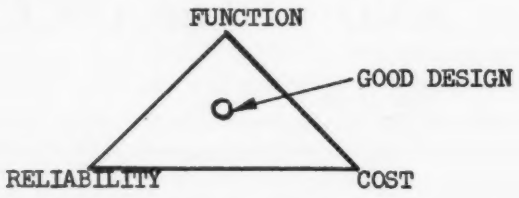
Once the designer develops the shape of his part as required by its usage, decides on the material to specify and specifies tolerances and finishes, he has pretty well set the price of his item, particularly with equipment purchased by the Government. The ultimate producer who is usually chosen by competitive bidding has very little room to use his initiative or imagination. Design, material requirements and everything else of pertinence are specified in detail. About all he has to play with is labor and overhead. Labor has become fairly standardized since the minimum wage laws, and as overhead is usually tied in directly with labor, he is pretty well strapped.

The designer usually selects his material for strength, durability, availability, and somewhere down the line is cost. As the design engineer's primary responsibility is to design a functional, reliable item, the cost becomes of secondary consideration. Even if he is conscious of cost, he usually has neither the time nor facilities to really determine the cost of his various alternatives. Practically no consideration is given to the cost of tolerances and finishes. And yet, these are two items where considerable savings can be achieved in almost every case.

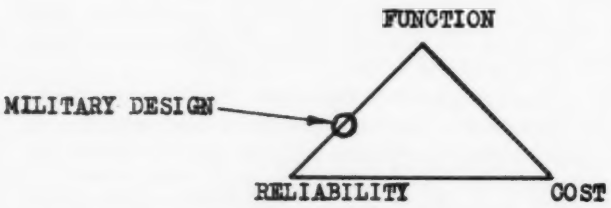
The designer has little more than common sense bolstered by limited experience and intuition to guide him in selecting finishes and tolerances required. And his judgment is many times overshadowed by his professional pride, which in an engineer makes him want to make his equipment, the results of his design, as nearly perfect as he can. In equipment this attitude is reflected in practically all specifications which carry the phrase, "best material and workmanship required." How much this phrase has cost over the years would be hard to imagine! Of course, in the interests of economy of material, machine and man hours and eventually money, we should design our equipment to be only as good as the job or function requires it to be. Anything beyond this is unnecessary waste! To quote from Doctor Vannevar Bush: "There is a common notion that during war costs do not count. There is no greater fallacy. The error comes from the belief that civilian resources are unlimited. They are not. . . . When war comes . . . only a limited amount of materials and manpower is available and the whole problem is to distribute these in optimum manner to secure the desired result."* It has been said that with enough

*From *Modern Arms and Free Men*, by Dr. Vannevar Bush.

time and money our American engineers can design anything. And again, any good engineer can reduce the cost of any piece of equipment by cheapening it, substituting materials, and in many other ways. But the real test of engineering at its finest is when the proper combination of function, reliability and cost is achieved. When equipment is designed to do its required job, no more and no less, perfection in engineering is achieved. This is very difficult to do and in an effort to make sure the function is performed or reliability achieved, the military engineer usually goes to the extreme and overlooks cost. This could be compared to a triangle with the apex of the angles being function, reliability and cost.



The proper combination of all three requirements would cause a design to be somewhat in the center. Too much emphasis on any one requirement detracts correspondingly from the other two. For example, an engineer's emphasis on reliability puts his design in a position like this:



In the automotive field, this blending of function and cost is probably more nearly achieved than in any other field of engineering. Batteries with 18 months' guarantee usually last just that long. A few give out sooner and a few last longer, but the majority last just that long. There are many other examples. But in the following paragraphs an attempt will be made to give examples of equipment that is considerably overdesigned and the resultant cost, as well as some guidelines in achieving the required function at minimum cost.

FINISHES

In the commercial field, a metal product can be satisfactory from a functional viewpoint, but unsatisfactory from an appearance standpoint. This can be a handicap in a competitive market where any slight advantage of one piece of equipment over another can mean the difference between making or losing a sale. This has resulted in many types of surface finishes, polished, plated, enameled and many others which serve no functional purpose, but do add to appearance and saleability. Unfortunately much of this has been carried over into the military field. In designing military equipment, appearance should play a very minor part. Any piece of metal which is polished or ground to a finish finer than that required by its use is an overdesigned and overexpensive piece of equipment.

At one time it was thought that fine surface finishes, in the neighborhood of 5-10 RMS micro-inches on marine reduction gears considerably reduced noise and added to the gear's load-carrying ability. Through controlled tests this was found to be not so, and it was found that the method of achieving surface finish was more important in determining load-carrying ability than was the finish. Charts 1 and 2 show the results of these tests. Accelerated endurance tests were run on a standard nickel moly steel to determine life of the material with various types and values of surface finishes. From Chart 2, it can be seen that ground and electropolished finishes lasted considerably longer than the other types tested. From Chart 2, it can also be seen that, contrary to previous indications, the coarser electropolished finish had greater endurance than the finer ground finishes.

This is somewhat borne out by the Chrysler Motor Company's experience with superfinishing. At one time superfinishing was specified for automotive piston skirts, brake drums, cam surfaces and tappet barrels and heads. These requirements grew out of laboratory tests, which showed possible increased performance. In actual service experience it was found that fine surfaces required a longer break-in time, sometimes longer than the allowable time, and parts were scuffed. When rougher surface finish was allowed, high initial wear followed which provided adequate smoothness for stable operation.

The same basic reasoning can be applied to tolerances on machined parts, drilled holes, etc. Any tolerance that is tighter than is required for the particular function of the part is a waste of man and machine time, and thus money. In every case that a tolerance can be increased without adversely affecting the product, it will result in a real decrease in manufacturing cost. It is always the last few increments which are the most costly, whether they are in tolerances, finishes or any other field. The smaller the margin of error tolerated, the more costly the operation.

In the competitive commercial field, extreme tolerances and finishes are often encountered. But

* From Modern Arms & Free Men by Dr. Vannevar Bush

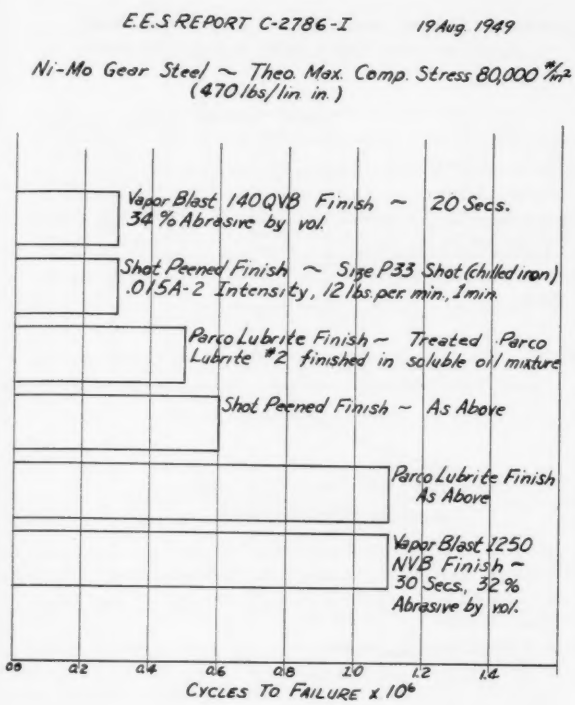

E.E.S REPORT C-2786-I 19 Aug. 1949
Ni-Mo Gear Steel ~ Theo. Max. Comp. Stress 80,000 #/in²
(470 lbs/lin. in.)
Vapor Blast 140QVB Finish ~ 20 Secs.
34 % Abrasive by vol.
Shot Peened Finish ~ Size P33 Shot (chilled iron)
.015A-2 Intensity, 12 lbs. per min., 1 min.
Parco Lubrite Finish ~ Treated Parco
Lubrite #2 finished in soluble oil mixture
Shot Peened Finish ~ As Above
Parco Lubrite Finish
As Above
Vapor Blast 1250
NVB Finish ~
30 Secs., 32 %
Abrasive by vol.
0.0 0.2 0.4 0.6 0.8 1.0 1.2 1.4
CYCLES TO FAILURE x 10⁶

Chart 1

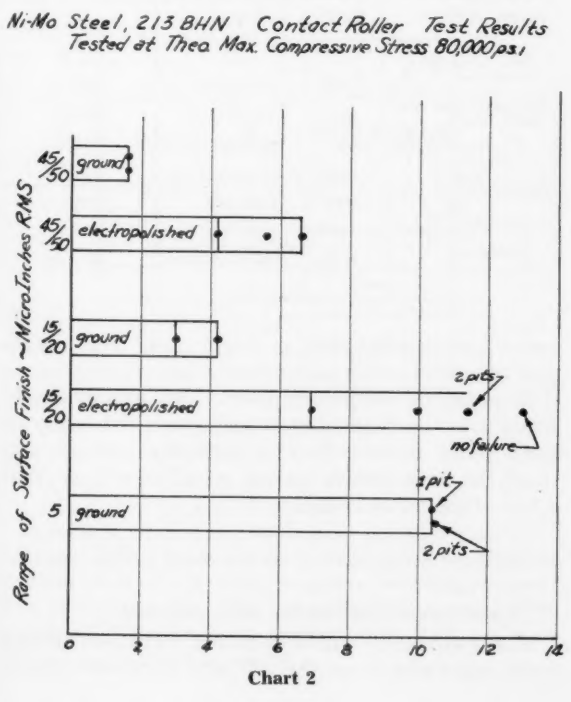

E.E.S. REPORT 4A(13)966904 13 AUGUST 1952
Ni-Mo Steel, 213 BHN Contact Roller Test Results
Tested at Theo. Max. Compressive Stress 80,000 psi
Range of Surface Finish ~ Micro Inches RMS
45/50 ground
45/50 electropolished
15/20 ground
15/20 electropolished
5 ground
2 pits
no failure
1 pit
2 pits
0 2 4 6 8 10 12 14

Chart 2

there, at least through the competitive process, the odds are that these areas will eventually be highlighted, evaluated as to cost versus function, and eliminated. In the field of military engineering, there is less likelihood. Once tolerances and finishes are put into a piece of equipment, and that equipment is working satisfactorily, there is a natural reluctance to change any part of it. After all, those who have been designers know many of the unforeseen and unknown factors which go to make a seemingly perfect design unworkable. So here once again, the factor of reliability comes into view to resist change and to perpetuate any overspecification.

There are many examples where sizeable savings have been achieved by recognizing these areas of overspecification and having them changed.

A. On a carrier of the Forrestal Class, there were hundreds of armored hatches which were ground smooth at a tapered surface and fitted to a sealing fit. The hatches didn't seal on the tapered surface but on the flat surface above. A burning cut would provide an acceptable surface finish for the function on the tapered surface. Something around 1000 micro inches. Yet few of the people involved were satisfied with less than a finished surface. Anything else is too expensive for the job.

B. In the installation of a reduction gear on a CVA, Naval Shipyard personnel proposed to re-assemble the unit in the shop instead of on board. There was considerable discussion with the manufacturer and the Bureau about the possibility of maintaining alignment. The requirements of plus or minus one one thousandth (±.001) on the shaft were quite extreme for service requirements. Then a mistake by the manufacturer was discovered which called for a re-alignment of more than ten thousandths (.010) of an inch. When it was found out this mistake would cost $2000, the shaft alignment was decided to be well within requirements. It is surprising how often putting a price tag on extreme requirements will assist in having them changed.

C. One more simple example. In a submarine, the Emergency Fresh Water Tanks were to be made of CRES steel sheet with a Number 4 surface finish, which cost 65¢ per lb. It was found that CRES steel with a Number 1 finish was all that was required for the job and cost 52¢ per lb. A savings of over $1000 was achieved, just by changing surface finish requirements.

Not only are tolerances of importance in the design stages, but they are also important in the interpretation of specifications. To illustrate: periodic inspection of anchor chains on all Navy vessels is required. Measurement for stretch and wear cannot be readily accomplished unless a vessel can avail itself of crane and pier facilities. On ships having a drydock period, the chain is laid out on the drydock floor, sandblasted, painted and measured. The Bureau of Ships standard drawing on anchor chains required that the overall length of six links be checked. A minus tolerance of 1/8" or a plus tolerance of 3/8" for each one-inch of wire diameter would be permitted. On this standard plan, in addition to tolerance, a maximum and a minimum dimension for a six-link shot of chain was given. Personnel involved inter-

Comparative Charts on Surface Finish

Surface Roughness in Micro-Inches	Approximate Relative Cost
500	100%
250	250%
125	500%
63	1100%
32	1800%
16	3500%
8	6000%

Symbol Designation	Comparative Micro-Inch Finish
Rough "f" (File or Grind)	500
Rough "f" (Machine)	250
"f" (Standard)	125
Fine "f"	63
Journal "f"	32
----	16
Lapp "f"	8

Chart 3

preted this to mean nominal dimensions plus allowable tolerance. Any section of chain having 6-link sections beyond this measurement was recommended by the Shipyard for survey. This was also used for the manufacture of new chain. Any chain measuring between the maximum and minimum tolerances was acceptable. A new chain near the maximum dimension would have little stretch before it would exceed the allowable tolerance. The purpose of all of this was to interpret chain strength by the amount of stretch. In this case when it was decided to add allowable tolerance to the maximum plan dimension, it was found there were 175 shots of chain within the new allowable tolerance that had been rejected the year before. These were reclaimed at a savings of $75,560. All in all, it was estimated in 1956 that a savings of $98,000 was achieved with the more liberal interpretation of the requirement. The chain was the same as before, and satisfactorily performed its function. Yet under one interpretation it was considered unsuitable.

When tolerances and finishes are once looked at with the view of what is the *maximum* tolerance and surface finish one can accept, rather than what is minimum one can specify, progress in cost reduction begins to be achieved. You will find you are getting

Comparative Cost on Various Methods of Machining Holes
(Based on Lowest Cost @ 100% - Quantity 200 Pieces)

Method	Lowest Tolerance Possible	Comparative Cost Tool Cost	Comparative Cost Bl. Cost*
Punch (Template)	+.004 -.002	100%	100%
Drill (Jig)	+.007 -.002	175%	300%
Drill & Ream (Jig + Bushings)	+.0006 -.0000	225%	400%
Bore (Fixture) Incl. punch, rough & finish bore on Borematic	+.0004 -.0000	540%	700%
Hone or Lap (Fixture) Incl. Punch & Bore	+.0002 -.0000	730%	1100%

* The setup costs, based on 200 plate quantity (.180" thick brass plate 3.50 x 5.00 size) have been included in the direct labor costs.

Chart 4

Cost Data on Machining Operations
(Based on Quantities of 1000 Pieces)

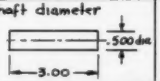
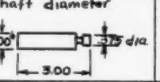
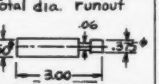
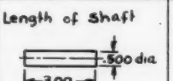
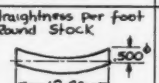
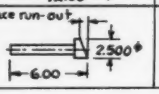
Typical Examples (Steel)	Tolerance	Operations	Approx. Cost Labor and Tools
Shaft diameter 	± .002 Stk ± .001 +.0000 -.0005	C/O Turn C/O Turn, C/O, Grind	100% 425% 460%
Shaft diameter 	± .002 Stk ± .001 +.0000 -.0005	Turn, Form, C/O Turn, Form, C/O Turn, Form & Grind	100% 130% 140%
Total dia. runout 	.005 .001 .0002	Turn, Form, C/O Turn, Form, C/O Turn, Form & Grind	100% 130% 165%
Length of Shaft 	± .02 ± .005 ± .001	C/O C/O Face, C/O	100% 110% 150%
Straightness per foot Round Stock 	.005 .001 .0002	Straighten Straighten + Grind Straighten + Grind	100% 375% 1250%
Face run-out 	.005 .001 .0005	Turn C/O Turn, Face, C/O Turn, Face, & Grind	100% 200% 400%

Chart 5

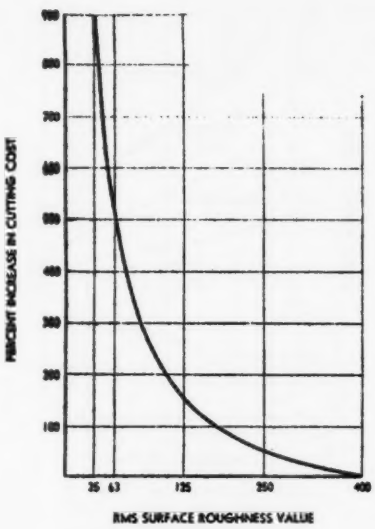
away from special parts in many cases. This means you are eliminating unnecessary variety and minor differences in component parts. This automatically leads into the field of standardization for many of these parts. Standardization is another subject and much has been written on it. It is sufficient to say that a few of the advantages are:

Fewer different parts to be purchased and stocked.

Greater production of standardized parts—less cost per part.

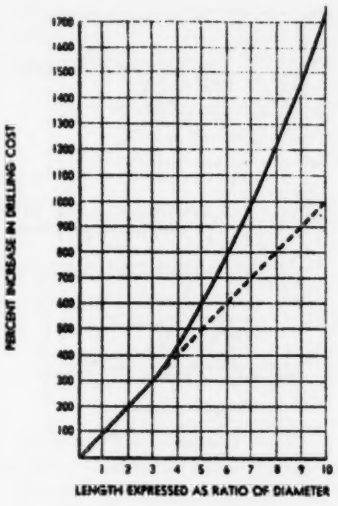
Less design time and drawings required.

There are many others of course, but these are the most important from the military engineer's view-



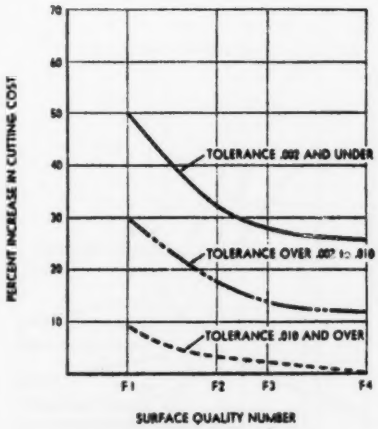
Cost Comparison Chart 1
Turning Cut
Varying Surface Quality

Figure 1. Shows relative cost comparison with respect to surface quality standards and limited to the turning process as done on turret lathes.



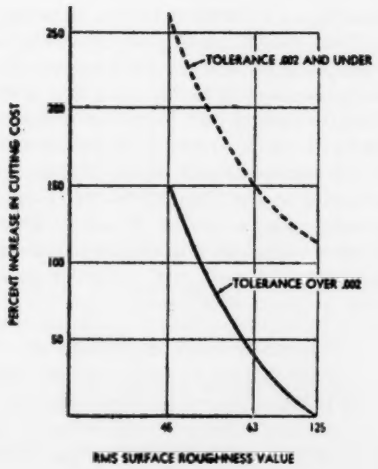
Cost Comparison Chart 2
Drilling
Varying Depth of Hole

Figure 2. Illustrates the cost behavior of drilled holes. It is readily apparent that deep holes are expensive. Beyond the depth of three times the diameter, the cost rises more rapidly than equal increments of depth.


PERCENT INCREASE IN CUTTING COST
TOLERANCE .002 AND UNDER
TOLERANCE OVER .002 to .010
TOLERANCE .010 AND OVER
F1 F2 F3 F4
SURFACE QUALITY NUMBER

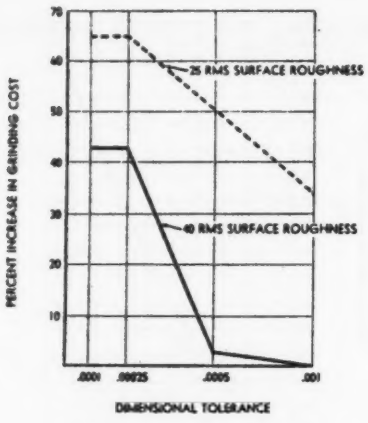
Cost Comparison Chart 3
Planing Cut
Varying Surface Quality
Varying Tolerance

Figure 3. Illustrates a cost behavior chart on planning, demonstrating the effect of two quality variables, one—surface quality and the other dimensional tolerance. Surface quality is defined in coded numbers since RMS values do not express the desired control.



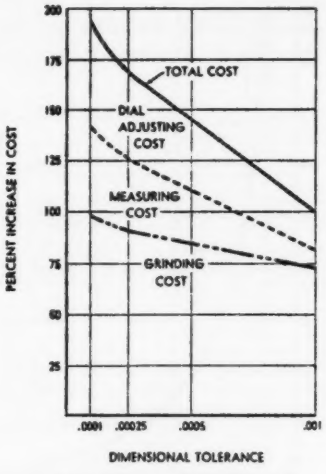
Cost Comparison Chart 4
Face Milling Cut
Varying Su-face Quality
Varying Tolerance

Figure 4. Illustrates cost behavior combining the same two quality variables as figure 3, but related to face milling.



Cost Comparison Chart 5
Cylindrical Grinding
Varying Surface Quality
Varying Tolerance

Figure 5. Shows cylindrical grinding cost, exposing a non-uniform cost behavior pattern.



Cost Comparison Chart 6
Internal Grinding
Varying Tolerance

Figure 6. Shows that reducing tolerance increases the cost of grinding, but also shows substantial cost increases occur from increased gaging and dial re-adjusting time required.

point. One other factor the military engineer should consider when he is in the design stage is whether or not there is a standard commercial part available that will perform the same function. If not exactly suitable, could it be modified at less expense than designing a new part? Or better still, could the military requirements be relaxed without affecting function, to use the commercial part?

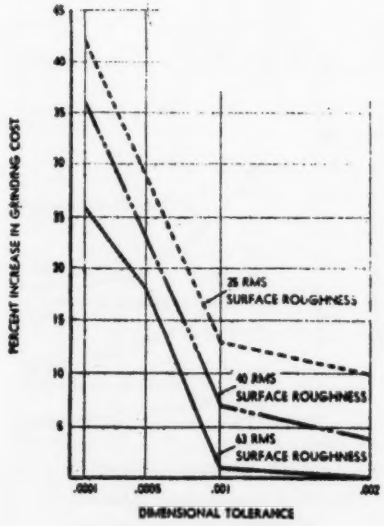
As an example, a military specification—MIL-T-18956 (SHIPS) of 4 October 1955 is in being on glass ash trays. This specification goes into detail on trueness and shape, color and marking required. In addition, a test procedure is set up for a 70° inclined test and a cigarette sliding out of its slot constitutes failure of the ash tray. A drawing is included of the ash tray with tolerances as well as of the test apparatus. These ash trays in the General Stores Supply Catalogue are carried at a cost of 30¢ each. The General Services Administration Catalogue carries a similar glass ash tray at 14¢ each, and a slightly smaller one, 4½" square, at a cost of 6¢ each.

SOME DESIGN CONDITIONS

Is the finish specified too extreme for the end use?

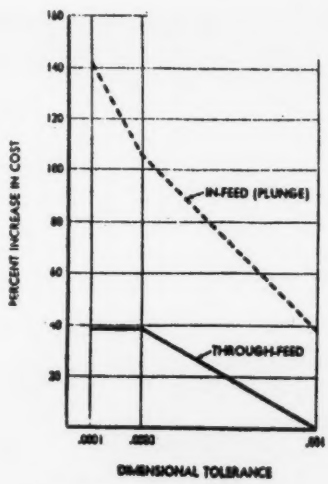
Mating metal parts are frequently machine finished. Could a throw-away rubber or other type of washer be used to get a seal? Example: A Philadelphia Naval Shipyard Value Engineering Project substituted throw-away rubber gaskets in submarine globe and angle valves in place of machined fit. Saves $99,360 yearly overhaul costs.

Is the finish specified too extreme for the production method? For example: many finishes are specified for castings, which are not only unnecessary from the use standpoint, but are very uneconomical in the production method. As a result much wastage of parts results, which



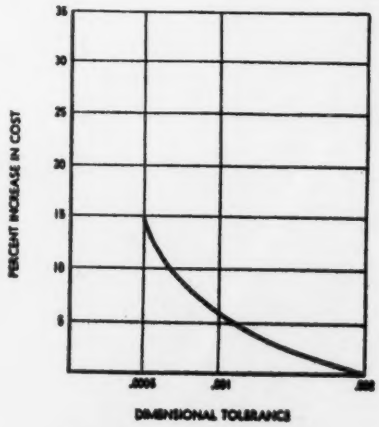
Cost Comparison Chart 7
Surface Grinding (Reciprocating)
Varying Tolerance
Varying Surface Quality

Figure 7. Illustrates that in reciprocating or surface grinding, the dimensional tolerance range has greater cost effect than the surface quality range. It is apparent that cost rises sharply with tolerances less than 0.001 in.



Cost Comparison Chart 8
Grinding (Centerless)
Varying Tolerance
Varying Design

Figure 8. Demonstrates, as does Figure 2, the possibilities of incorporating quality features other than surface roughness or tolerance. The difference between the two curves is governed by the design of the part. In order to gain the benefit of the low-cost "through-feed" method on the centerless grinder, the ground diameter must be the largest dimension on the part. Should such design be impracticable, the engineer is guided toward selection of a generous tolerance for the resulting "in-feed" method in order to minimize cost.



Cost Comparison Chart 9
Surface Grinding (Rotary)
Varying Tolerance

Figure 9. Covers rotary surface grinding and demonstrates primarily that not in all cases is cost severely affected by quality variation. The full cost increase over the whole tolerance range is 15 percent and is by far the smallest of all the illustrations. Yet even here it is seen that more than half the cost increase occurs between the 0.001 and 0.0005 tolerances.

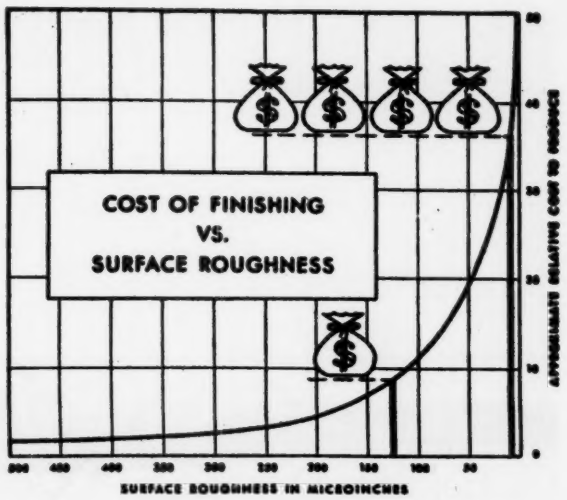


Figure 10. General Chart on Surface Roughness versus Cost.

the customer, of course, pays for. Consult the casting experts before specifying finishes and tolerances.

Are tolerances specified practical and economical in view of end use?

Many times in the past, on catapults, 12 sections, 40 feet in length, were machined with extreme accuracy, then installed and pulled out of shape by welding them together.

Is the design restricted to one production method only, or is it flexible enough to lend itself to several methods enabling the most economical method to be used?

One possible way of emphasizing the cost of finishes would be to have comparative surface finish blocks available to design engineers and shop personnel. In addition, charts should be made up to accompany these blocks showing the comparative costs of the different surface finishes. This would give those concerned a more realistic view of finish requirements vs cost.

And finally to repeat, are all finishes and tolerances specified to the maximum *allowed by the functional use?*

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CHARTS

Charts 1 and 2—Engineering Experiment Station, Annapolis Credit

Charts 3, 4 and 5—RCA Corporation Shop Practice Credit

Figures 1 through 9—From Reference Number 1

Official U. S. Navy Photograph

USS PARSONS, DD949, takes to the water on August 19 from the building ways of Ingalls Shipbuilding Corporation. The most recently launched of the DD931 Class, the ship will be ready to take its place in the fleet in a few months.

SEMICONDUCTORS . . . FROM CRYSTALS TO DEVICES

ACKNOWLEDGEMENT

This article was written by L. L. Friend, Manager, Manufacturing Semiconductor Department, Westinghouse Electric Corporation, Youngwood, Pa. It was published in the July 1958 "Westinghouse Engineer."

THE SUCCESS of a semiconductor device depends to an unusual degree on painstaking care in the preparation of basic ingredients, and equal care in the manufacture of the devices themselves. For example, some materials must have one part impurity per several billion parts of semiconductor. In the fabrication of devices, the demands are not quite this stringent, but nonetheless extremely exacting. Consider, for example, some of the operations in the recently completed Westinghouse semiconductor plant.

Here the production of semiconductor devices is divided into functional areas, covering the operation from crystal pulling to completed devices. These include:

Single-crystal pulling area
Crystal slicing and preparation area
Silicon-diode fusion area
Low power silicon-diode line
Medium power rectifier assembly line
High power silicon-diode encapsulation line
Rectifier bridge assembly area
Audio transistor area
Hyper-pure silicon production area

SINGLE-CRYSTAL PULLING

One of the most exacting and difficult operations of semi-conductor manufacture is the production of high-quality, high-yield, single-crystal material for subsequent device fabrication. Both germanium and silicon polycrystalline materials are "pulled" into single crystals in furnaces under either inert-gas or vacuum atmospheres.

Inert-gas furnaces predominate since most devices manufactured use n-type crystals, which must be pulled in an inert atmosphere such as argon.

The essential features of a typical gas-pulling unit are the crucible assembly, the pulling head assembly, power supply, and controls. The operating cycle is something like this: For silicon single-crystal pulling, a cylindrical flat-bottomed quartz crucible is used to hold an initial silicon charge of 90 grams. The crucible nests inside a cylindrical graphite crucible holder, which in turn is surrounded by a spirally-wound, water-cooled, high-frequency copper heating coil. This crucible holder and support assembly is designed to rotate counter-clockwise at 10 to 30 rpm.

A water-cooled crystal-pulling rod assembly is suspended vertically above the rotating crucible. In operation, a two-inch-long "seed" of previously pulled single-crystal material is attached to one end of the pulling rod yoke, and this entire assembly can be rotated clockwise from 4 to 100 rpm, and moved vertically from 1.5 to 6 inches per hour. A charge of polycrystalline material is loaded in the quartz crucible and an n-type doping material is added. Then, the entire crucible and pulling-rod sections are assembled and isolated from the surrounding atmosphere by a quartz furnace tube about 3½ inches in diameter and 21 inches long.

To pull a single crystal, the material in the quartz crucible is brought to temperature: 1450° C for silicon, 950° C for germanium. Next, the pulling rod is lowered until the end of the seed contacts the melt and rotation of both seed and crucible is started. As

soon as "points" form at the end of the seed, the seed is slowly withdrawn and the single crystal is pulled to completion, with rotation and withdrawal accomplished at a programmed rate. Temperature control is maintained during the pulling period within 0.1° C of the desired melt temperature.

For pulling of p-type single crystal for transistor devices, a vacuum atmosphere is used. The furnace construction is similar to that of the inert gas-pulling type, except that the charge is placed in a water-cooled metallic enclosure.

CRYSTAL SLICING AND PREPARATION

Silicon and germanium single crystals arrive at this stage in the form of bar stock, and although the material is a single crystal, it is not uniform in its electrical properties along the length of the bar. Therefore, the bar is tested and marked along its circumference into desired resistance categories. The slices taken from each section are separated for use in particular devices to take proper advantage of the resistivity characteristics.

Once marked, the single-crystal bar is mounted on a steel rod and inserted into the chuck of the slicing machine. A diamond cutting wheel then slices the single crystal into wafers automatically, moving the bar stock up slightly after every slice.

The final single-crystal preparation consists of cutting circular silicon "dice" of exactly the right diameter desired. Common dice diameters range from ⅛ to ¾ inch. This dicing operation consists of ultrasonically grinding the circular shape from the original wafer.

Again, sample slices of the material from any given single crystal are tested for electrical characteristics to determine suitability for particular devices. The cut dice are then washed with solvents, dried, and checked for proper thickness.

The slicing operation for germanium single crystals is similar to that performed on silicon. However, since all germanium crystal produced is used for transistor devices, surface damage—due to cutting—must be removed from the slices by machine lapping. This extremely sensitive operation automatically gauges and controls slice thickness, distribution of grit slurry, and cycle time. The lapped slices are then size-etched in acid to the required thickness, rinsed, dried, and inspected.

In contrast to the round silicon dice used for diodes, germanium transistor dice are square in shape. Silicon carbide grinding wheels cut the circular slices into dice for audio transistor use.

SILICON-DIODE FUSION

For practical use in semiconductor diodes the silicon dice must first be fused with counter-electrode and ohmic contacts. This procedure involves making a "sandwich" of a base molybdenum wafer, an ohmic solder disc, the silicon disc, a counter-electrode solder disc, and another molybdenum wafer.

These five-piece assemblies are loaded in proper sequence into cylindrical holes drilled in graphite fusion boats. Once loaded, small stainless-steel weights are placed on top of each sandwich to give the right amount of compression during fusion. Then the boats, 18 inches long and half-cylindrical in cross section, are loaded into a horizontal tube vacuum fusion furnace. Each boat is fitted with a thermocouple to measure boat temperatures during fusion. The fusion of the parts is performed by first heating rapidly to the required peak temperature and then cooling rapidly.

The fused diodes are then unloaded from the graphite boats and tested for forward and reverse voltage characteristics. The sandwich is finally tinned prior to encapsulation.

LOW POWER (50 MIL) SILICON DIODES

This line produces three types of rectifiers—the 307, 320, and the tv and radio rectifier, the 1N1169. The first two cover peak inverse voltage ratings from 50 to 1000 volts.

The encapsulation is a straight-line operation with all sequential welding and testing operations performed in an elongated dry box. The line is equipped with elastic-sleeved arm openings at each station through which the operator can work with ample freedom of movement. The Lucite front panels of this enclosure permit full view of each work station. Air humidity within the enclosure is maintained below a dew point of —30° C.

Fused diodes are delivered to the first station of this line for testing. Diodes accepted for further processing in this line must have a satisfactory reverse leakage at 150° C.

The trays are then moved to the internal spring weld station where a flexible conducting spring is welded to the top of the tantalum nail shank that serves as a base for each diode. From here the diodes are transferred to the next station, the external "top hat" project-welding machine. The "top hat" portion of the case is cylindrical in shape and has a circular welding flange at the bottom and a compression glass seal at the top through which a thin hollow tube extends.

In this station, the primary encapsulation is completed by fitting the spring lead of the diode inside the "top hat" portion of the case and welding the flange to the molybdenum diode base. Several units of each lot completed here are removed to a leak detection station for leak testing.

Lots from this operation are then loaded in a vacuum where they are pumped down and backfilled with dry air. This is called a vacuum-flushing operation. "Flushed" diodes are then moved to a station where the thin hollow tube in each diode is crimped to hermetically seal each unit.

After other quality control tests to determine suitable electrical characteristics of each device, they are separated into various voltage classifications by a series of hot (150° C) and cold (25° C) tests. Classification is done on semi-automatic machines known as "pin ball" testers. Their name is derived from lights that flash "on" in sequence during tests and indicate the voltage classification of each device.

MEDIUM-POWER RECTIFIER ASSEMBLY

This line produces four types of rectifiers as standard product with peak inverse voltage ratings from 50 to 600 volts maximum at peak inverse current of 10 and 20 ma. However, many variations of the standard types are also processed for special order.

The encapsulation line is also a straight-line operation of elongated dry box construction similar to the 50-mil diode line. At the beginning of this line, the top header and spring assemblies are made.

The top header is a steel cylindrical member with a compression-type glass seal at one end, which supports a copper-plated alloy steel centerpost. A copper spring is soldered to the end of the centerpost.

Next, at the staking station, all the cell components are mechanically assembled. Here a screw-machined copper stud-type base, a pre-tinned silicon diode, a mica insulation washer, and the top header assembly are staked together by an air-operated staking tool and fixture. A solder ring is next lowered down around the top header until it rests snugly against the shoulder of the stud case.

A number of these rectifier assemblies are loaded onto an aluminum soldering tray and placed on a hot plate and covered by a transparent Pyrex glass. During the soldering operation, a nitrogen-hydrogen mixture is introduced under the Pyrex cover. A simple electrical test is then made to discard all "shorted" or "open" cells.

Subsequently, all units are leak tested, buffed, marked, nickel-plated, and inspected. Both a high-temperature (150° C) classification and room-temperature classification are made. A cell is always classified at the lower rating registered in each test.

HIGH-POWER SILICON-DIODE ENCAPSULATION

This line produces five types of rectifiers with peak inverse voltage ratings from 50 to 500 volts maximum at peak reverse currents of 40 and 50 ma maximum. These devices use the standard ½ and ⅝-inch diodes. Again, the enclosed dry-box production method is employed.

Again, at the beginning of the line, the top header and internal spring assemblies are made. The difference, however, is that for the high-power diodes, the compression spring is made of stainless steel.

The general scheme of operations is similar to those used in the medium-power diode assembly.

After all testing of these units has been completed, an external tinned-copper, flexible braided lead with spade terminal is swaged into the outer socket of the cell centerpost.

RECTIFIER BRIDGE ASSEMBLY

A wide variety of bridge assemblies—mounted combinations of rectifier cells—is produced. For this reason, and because of the low volume associated with any particular type, little has been done to adopt mass assembly methods and techniques in this area. At present all bridges are hand assembled and hand soldered on work benches arranged for convenience of work flow. In general, assembly efficiency has been increased by the use of power-driven hand tools, and where volume has warranted, by relatively simple fixturing.

Typical units assembled here include single-phase battery charger bridges, three-phase welder bridges, and three-phase bridges used as replacements for motor-generator sets in some volume applications.

TYPE 100 (AUDIO) TRANSISTORS

This production area is a product line in the final phase of manufacturing development. This means that each station, in manufacturing limited quantities of product, is being studied in detail for methods improvements. The present manufacturing operations are carried out in a totally enclosed "dry-room" area with every attempt made to maintain hyper-clean conditions.

Germanium dice are cleaned and dried before shipping to this area. Then the dice, tin-coated nickel-base tabs, indium emitter and collector pellets, and the collector and emitter lead wires are assembled at a fusion-boat loading station. Fusion of the complete assembly is then carried out in a special two-piece graphite fusion boat.

After the fused assemblies pass an electrical test, they are welded to a base tab and header. As can be imagined, these welding operations are extremely precise and must be performed by tweezer-weld machines with special holding fixtures and welding electrodes. After more testing and cleaning processes, the units are now ready for final capping, painting, and shipment.

HYPER-PURE SILICON MATERIAL PRODUCTION

Until recently, available silicon has left something to be desired in the way of quality.

Manufacturers of devices have found that this silicon, while pure indeed, is not pure enough. Using the best available material, fused large-power diodes can be made with ratings to 500-volt peak inverse voltage. Purity of silicon had to be improved so that the peak reverse voltage characteristic would range to 1000 volts.

The Siemens process as recently developed by the Siemens Schuckertwerke and Siemens Halske in Germany, in conjunction with work done by Westinghouse, has produced silicon material to satisfy the above requirements. Through an agreement

with Siemens, Westinghouse has licensed this process in the United States.

A pilot facility has been installed for working out quality and yield improvement problems of this process.

Because of the exacting standard necessary to achieve desired characteristics from semiconductor devices, production facilities are unique. As the industry grows, many changes will undoubtedly take place in the facilities as a result of new technology. Full provision is made for such flexibility in this new plant.

A new refractory material with striking properties has been produced by the Pittsburgh Corning Corporation by causing silica to boil, foam and harden. It weighs 10 to 12 pounds per cubic foot, and has a small but positive coefficient of expansion up to 2200°F. The insulating value of 1 inch of this product, called "Foamsil", is said to be equivalent to 18 inches of acid brick at 250°F. In addition to other remarkable properties, the substance is highly acid resistant and non-absorbent.

—from MECHANICAL ENGINEER
—September, 1958

To improve the control of large tankers in restricted waters, a bow propeller installation has been proposed by Mr. E. Goldsworthy. The installation consists of a cycloidal propeller, mounted with its axis vertical, located in the forward peak tank so that the propeller projects into an athwartship tunnel. By control of the propeller blade pitch, a jet can be directed out either end of the tunnel, providing a sidewise thrust at the bow which would be highly effective for maneuvering in close quarters. It appears possible that a direct financial advantage would accrue by reduction of the number of tugs necessary to assist the ship in harbor waters. The cycloidal propeller is uniquely adapted to this application, since it can be run at constant speed and since the direction of thrust can be quickly changed. Additionally, the mounting of this propeller within the tunnel is accomplished with no brackets, bosses, or other protrusions into the water flow.

—from TRANSACTIONS OF THE INSTITUTE OF MARINE ENGINEERS, September, 1958

COMMANDER BERNARD M. KASSELL, U.S. NAVY

MARINE ENGINEERING NOTES FROM THE SOVIET PRESS

THE AUTHOR

enlisted in the Navy in 1931, and has served continuously since that time. He attended Cornell University and Maryland University, obtaining his B.S. degree in Military Science from the latter institute. He has had a wide range of duty assignments, including tours as Engineer Officer and Repair Officer on various ships and tenders. He is currently stationed in Casablanca, Morocco as Commanding Officer Military Sea Transport Service Office, Morocco. The materials contained in this article have been taken from Russian language publications. The translations are the work of the author.

UNDERWATER LABORATORY

THE ABOVE title appeared over a recent article in "Sovetskiy Flot," organ of the Red Navy, and concerned itself with the use of submarines for scientific research. According to the writer, V. Azhazha, who is attached to the All-Union Scientific-Research Institute for Marine Fish Life and Oceanography (VNIRO), work on the conversion of an unspecified class of Soviet submarine into an underwater laboratory was far enough along to anticipate the departure of a group of scientists in the submarine on an expedition last summer.

Soviet scientists want answers only to such questions as the stocks of fish and animal life in the seas, the factors of temperature, salinity, and the like as they pertain to fish life, resources of the seas' bottom, and the manners in which the riches of the seas can be better utilized. We suppose that any military information obtained will, since the Soviet Union is engaged solely in peaceful pursuits, be disregarded! Needless to say, the use of a vessel such as this can contribute a considerable store of information to projects concerned with such war-like pursuits as anti-submarine warfare, mine warfare and related subjects.

Initially the designers planned on installing ports in the forward part of the boat, with illumination to be provided by powerful searchlights. The scientists were to be comfortably seated in front of the ports and observations of external phenomena, as well as of the instruments installed within the hull, could thus be recorded. When the more practical engineers were confronted with the problem, however, it was found that the installation of ports was not so simple a matter as had first been supposed. First of all, no adequate expanse of bulkhead could be found amid the usual maze of piping, cabling and vent ducting found on board a submarine, which would lend itself to the cutting required. And if such a bulkhead could be found, the second problem, that of cutting through the heavy pressure hull and the light outer hull, and connecting the two, was not readily handled. The difficulty appears to have been overcome by the installation in the bow of television cameras which will record what they see on the screens of receivers inside the pressure hull.

In addition, the boat is to be equipped with sonar gear, fathometers, and means for taking water and bottom samples. A remote thermocouple will be installed so that an electronic indicator will record

temperatures and salinity of the water automatically at any depth and at any speed. Yet another indicator will be used to obtain oxygen content of the water, and current direction and velocity will also be recorded.

The experience gained by the first expedition will enable designers to plan special submarines for industrial and scientific purposes and, as experiments with atomic energy go forward, Azhazha notes, the very real possibility develops that atomic energy propulsion plants will be installed in future submarines. The use of such propulsion plants will expand very considerably the use of boats so equipped for scientific expeditions, in both the Arctic and Antarctic; and we might add, for military purposes as well.

Soviet writers have, of course, followed events with regard to United States submarines equipped with atomic energy power plants and, in general, have confined their remarks on the subject to straight reporting, including all available technical details and operating characteristics. An occasional report has been noted which follows the "peaceful" propaganda line in condemning the use of atomic energy for warlike purposes. At the same time, the present writer has, until the above mention was found, been unable to find anything suggesting plans for installation of an atomic power plant in a submarine. This is not to say, of course, that the possibility does not, or has not, existed; it is merely to say that, as usual, secrets are pretty well kept in the Soviet Union, particularly in the military field.

RAISING OF *Lepsye*

The case of the motorship *Lepsye* is curious indeed. Bare mention of the vessel occurred in mid-1956. In the spring of 1957 it was noted that *Lepsye* was an unfinished dry-cargo vessel which had been sunk in the Black Sea by German submarines during World War II. So far as can be determined, the hulk was lifted by the Soviet salvage organization, EPRON, and taken into the Novorossiysk Shipyard for inspection. At this time it was found that the bottom was deformed and that complete machinery replacement was necessary. Novorossiysk proceeded to prefabricate 37 sections of the bottom, upon completion of which work the vessel was docked, the bad sections cut out and the newly fabricated sections set in place. The work appears to have been completed in April 1957 and nothing more was heard of the 3000-ton displacement vessel until February of 1958 when it was announced that the hull, weighing 800-900 tons, was being put on the inclined ways in the Admiralty Yard in Leningrad. It is claimed that this is the first time in the history of shipbuilding that a ship of this weight has been towed up an inclined, longitudinal building way. Soviet experience appears to have been limited to lifts of 500-tons but in this case experts from the Admiralty Yard worked out the details and arrived at a solution which resulted in the vessel being drawn on to the slipway at a rate of 20-meters per hour using winches and cables. The end result of the use of a longitudinal slipway instead of a drydock is that, at the top of the slipway the vessel is well out of the way of ordinary work, a drydock is freed for regular use, and the slipway itself, at its lower end, still is available for building purposes. The *Lepsye* has the following characteristics:

Length	282 feet
Beam	40 feet
Height of side	25 feet

SMALL RIVER CRAFT

In the latter part of March V. N. Karpov, Chief of the Main Administration for River Transportation (GLAVRECHTRANS), revealed the scope of the 1958 building program for small craft and, at the same time, indicated that the small rivers will carry more agricultural products this year than was the case in 1957. In order to carry the additional load the Barnaul Shipyard was ordered to build 50 barges with a total carrying capacity of 7,500 metric tons for the Altay Kray alone. The Krasnoyarsk Shipyard is scheduled to produce 17 water-jet tug-cutters. Then, in order to bring grain out from the various collection points well upstream along the small Siberian rivers, the Kashira Shipyard, as well as Krasnoyarsk, were ordered to build a number of new type craft. Twenty shallow draft barges were also ordered from the Novosibirsk and Omsk administration shipyards and yet other craft were expected from shipyards in Tomsk Oblast.

As usual, passenger transportation creates its own problems, and in order to cope with what is announced as "increased" travel, the Krasnoyarsk Shipyard must also build 25 passenger cutters, each capable of carrying from 65 to 80 passengers. The cutter, somewhat similar to the *Moskvich* in design, will appear on the small rivers for the first time. Since they will draw very little water, they will be able to penetrate well upstream where, in addition to passengers, they will be able to pick up and deliver small lots of cargo.

THE HYDROFOIL MOTORSHIP *Raketa*

In 1957 the Soviet Union put a river passenger vessel called *Raketa* into service in order to determine if the design developed by Chief Designer R. E. Alekseyev of the "Krasnoye Sormovo" Shipyard was a valid one for future development. *Raketa*, as reported in the JOURNAL, November 1957, p. 725, is a hydrofoil craft which is stated to have made 39.1 knots in calm waters.

The principle of the hydrofoil is, of course, not new, but it may be of interest to the American builders to learn of the results obtained from operational tests made by Soviet engineers with their version of the hydrofoil.

The principle characteristics released by the Soviet press for *Raketa* include the following:

Length overall	88.3 feet
Length between perpendiculars	79.4 feet
Beam (not including rubbing strakes)	14.4 feet
Height of hull, including wheelhouse	14.8 feet
Displacement, loaded	23.0 tons, metric
Displacement, empty	17.8 tons
Draft, full load, not underway	5.9 feet
Draft, full load, underway, on foils	3.6 feet
Transverse metacentric height at full displacement	14.1 feet
Maximum speed, still water	39.1 knots
Minimum speed on the foils	23.4 knots
Cruising radius at 60-km/hour, maximum	500 miles
Passenger capacity	66 persons
Crew	4 persons

Propulsion is provided by one main engine, Mark M-811, rated at 900-horsepower maximum, at 1650-rpm. When the engine reaches 1200-rpm, according to recorded data, a point when the vessel begins to rise on the foils, fuel consumption drops off until the vessel is full on to the foils, 1300-rpm, at which point consumption rises and continues to maximum, 370 pounds/hour at 1650-rpm.

The engine is connected directly to the propulsion shaft through a reverse clutching mechanism. The engine may be started and controlled from both the engine room and the wheel house. The propeller originally fitted is three-bladed, of welded construction, made of stainless steel, 2.2 feet in diameter, pitch 2.9 feet, and a disc relationship of unity. The drive shafting, also of stainless steel, lies in two rubber bearings at an angle of 12 degrees to the plane of the waterline. Additional equipment in the engine room includes a Type GSK-1500 generator, coupled to the main engine, an auxiliary compound generator Type KG-5.6, and two acid storage batteries Type 6STK-180. The main lighting is a two wire, 24-volt system.

All this equipment is fitted into a two-decked hull built of riveted duraluminum. The bow is spoon-shaped, the stern square. The passenger compartment is located between frames 2 and 28. Within this space are fitted 66 aircraft-type reclining seats, as well as a small buffet. Ten to twelve additional persons can be accommodated in a covered stern area. The machinery mentioned earlier is located between frames 28 and 39, and from frame 39 to

Figure 1. The Motorship RAKETA Underway.

Figure 2. General View of the Passenger Compartment.

frame 42 is the head and washroom area. Anchor and mooring gear is found in the covered bow area.

There are two sets of hydrofoils fitted to *Raketa*. The bow set is arrow-shaped and is in the form of a V, while the stern set is rectangular. Each foil is secured to the hull by three struts, the propeller shaft housing forming the amidships support for the after foil. The struts are made in two parts and are not interconnected. The lower part of the strut is welded to the surface of the foil, while the upper part is fastened to the hull with bolts. Both parts of the struts are flanged and are fastened together by bolting. Special quoins, located between the flanges of the struts, can change the installed angle of attack of the foils which, in profile, are flat-convex. The foils are made of welded stainless steel plating and the leading edges are sharp.

A special stabilizer, or sub-foil, is located directly abaft the outboard struts of the bow foil. The stabilizer is considered to be a part of the hydrofoil installation and its purpose is described as assuring the transverse stability of the vessel during the transition period (hull to foil movement), and to limit the list of the vessel as it rises on the foils, as well as when running on the foils.

The rudder is of ordinary design, welded, with a total area of 13 sq. ft., 7.7 sq. ft. of which attack the water when the vessel is on the foils. Steering is accomplished by hand through tiller ropes connecting the wheel with the rudder.

Raketa, since it is beyond the design stage, provides a few interesting footnotes to hydrofoil operations. Although the vessel was assigned to the Volga Combined Steamship Line for operational use, the assignment was, in fact, a means of deriving useful data for future construction of the series, said to be contemplated in several shipyards within the Soviet Union. Nevertheless, *Raketa*, between 24 August and 14 October, 1957, is said to have piled up some 12,000 miles of operations between the cities of Gorkiy and Kazan, a distance of 250 miles. Scheduled time for the downstream run was eight hours, that for the return run thirty minutes longer. Included

in the schedule were four, ten minute stops at intermediate points. By way of comparison, the usual fast Soviet passenger ship on the same run is scheduled for 23.5 hours downstream and 30 hours in the upstream direction. The runs, evidently, were not without incidents. *Raketa* is said to have gone aground twice while running at full speed. In each case the vessel was towed off the shoal by a tug and made port under her own power, running on the foils in each case. Inspections made at a later date revealed no damage. In rough water, with wave heights a reported one meter, *Raketa* averaged 27 to 30 knots and experienced only occasional jolts from the waves. Toward the end of the season, when only the largest passenger vessels were still in service, *Raketa* maintained the set schedule.

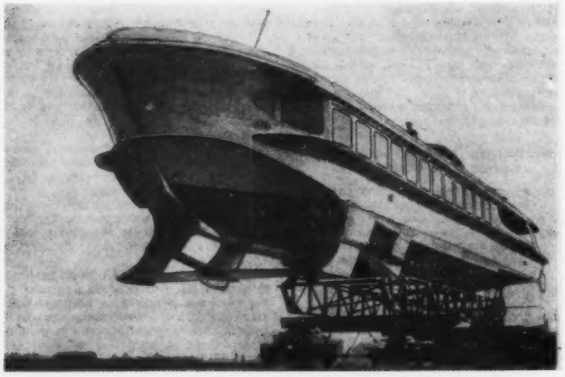


Figure 3. General View of the Underwater Body and the Hydrofoil Installation of the Motorship RAKETA.

What does the future hold for the *Raketa* design in the Soviet Union? According to Engineer I. M. Kolyishkin, the Soviet Union can expect big things from the design, provided certain defects which appeared can be eliminated. The propeller and shafting installed in *Raketa* proved to be of poor design, certain of the rudder parts did not stand up under operational use, and, a minor point, the rubbing strakes will have to be replaced by larger and wider pieces. Certain minor defects within the hull structure also require elimination. For the future, the demonstrated comfort and stability of *Raketa* make it completely feasible, from the Soviet point of view, to attempt to design larger passenger ships using the hydrofoil principle, not only for the rivers, but for coastal operations as well. This, admittedly, presupposes that some way is found to overcome the defects in design, particularly in the propeller and shafting, already noted.

THE SEVEN-YEAR PLAN FOR THE MERCHANT FLEET

In the course of the April All-Union Convention of Maritime Transportation Workers, Deputy Minister Kolesnichenko pointed out that the Soviet Union is faced with increased passenger travel. Thus, the fleet is in need of new ships and, within the framework of the Seven-Year Plan ending in 1965, plans have been made to build several series of ships. Included in the plan are large passenger ships for the Far East, as well as for foreign runs, carrying from 700 to 1,000 persons each; a class for the Black Sea with a 300 to 500 passenger capacity; 240 passenger ships for the Caspian and Azov basins; a class for use in northern waters; and cutter types for local runs. Included is a series of ten hydrofoil craft of the *Raketa*-class, as well as four ferry boats for the Caspian which will carry trains as well as passengers. Not all of this production will come from Soviet shipyards, however. A series is already under construction in Wismar, East Germany. The first of the class, *Mikhail Kalinin*, brought forth critical comment from the Deputy Minister, who objected to finding de luxe cabins full of noisy piping, while first class cabins are equipped with tier-type bunks rather than beds. The vessel will be assigned to the Leningrad-London run when it is placed in service.

ELECTRIC DRIVE VS DIESEL

Not all Soviet marine engineers are agreed on the mode of propulsion best suited to the needs of the country, according to diesel advocate Engineer M. A. Kharitonov. Despite the fact that, in the past five years, the Soviet river fleet has been adding diesel-electrics to its lists, the argument is still unsettled. Kharitonov is not convinced that the operation of these vessels has settled the matter. In fact, many mechanics and engineers feel that diesel drive is far superior to diesel-electric, particularly as installed in the new 800-hp passenger vessels coming out of Czechoslovakia. No one, it seems, has bothered to cast a critical eye on these diesel-electrics which are, actually, full of defects.

The main fault to be found with electric drive, according to this claimant, is that there is a built-in 18 percent power loss. This is to say that, in the class cited, despite the 800-horsepower rating, there are actually uncompensated losses of 144-horsepower. On the 36 ships of this series, therefore, the total power loss is on the order of 5,200-horsepower, an amount equal to the power of five Volga passenger ships such as *Volodarskiy, Polkovodets Suvorov* and others.

Nor is this all. There is a vast difference in the building costs for the two types of drives. According to data provided by the building yards, the cost of the electrical installation in a diesel-electric is an average 20 to 23 percent of the total cost of the vessel, whereas the cost of the electrical installation in a diesel ship is only 5 to 7 percent of such cost. In operation, a diesel or steam propelled river ship usually carries but one electrician, but the diesel-electric requires three to four trained electrician-mechanics.

Maintenance costs for a diesel-electric exceed those for a diesel, not only because of the added manpower and the burden of repairing the electrical equipment, but because few yards in the Volga River basin can do the repairs to the equipment installed in the electric drive vessels.

Nor are they money-makers. When the season is completed they are laid up in wintering places, idle. At this point it can be noted that the argument is certainly weak, for all Soviet river ships winter when the ice sets in. However, objections based on low backing power, difficult hand manipulation of controls, the great length of time taken to launch boats (which is undoubtedly due to a design feature), inadequate ventilation and other, unspecified defects, are somewhat more valid.

DIESEL ENGINES

The model 8DR engine, built in quantity during the period 1952-1955 in the "Russkiy Dizel" Works for the *Kazbek*-class tankers is, from all reports, badly designed. Operating difficulties necessitated the convocation of interested parties as far back as 1955 in order to determine what could be done about improving the design of the engine. In the intervening three years some improvements have appeared, the result of efforts on the part of the engine builders, as well as from recommendations made by operating personnel.

In 1956 the engine builders designed a new injector for the models D and DR engines. Extensive tests produced satisfactory results and, at the present time, the new model injector is in serial production. The most important individual components, such as sprayer plates, needles, nose washers, springs, push rods and connecting fittings, are in mass production in plants throughout the Soviet Union, the result of standardization of such parts. The changes made for incorporation in the new model injector include a change in the angle at which the holes in the sprayer plate are drilled, as well as a decrease in the diameter of the holes, a popping pressure that is somewhat lower than previously, and moving parts that are lighter in weight. It is claimed that with use of the new model, fuel consumption has been reduced, the saving ranging from 12 down to 7 grams per effective horsepower per hour. Manufacturing costs have dropped 78 rubles per unit.

Other problems remain. Early in 1958 a decision was reached to put the same engine back into serial production, but this time incorporating in the design the various changes and improvements which operations seem to dictate. This decision was taken purely for reasons of expediency, for it is admitted that the improvements have done little to increase power, provide greater economy or increase reliability. By June 1958, evidently because of continuing dissatisfaction with the engine, yet another conference was convened and was attended by all the interested parties. Chief Engineer Yeromoshkin, of the Black Sea Steamship Company, operator of all *Kazbek*-class tankers in the merchant fleet, made the most important speech. Based on the facts which he presented on the subject of operational faults in the engine, the conference decided to make concrete proposals to the engine builders in a further effort to improve characteristics of the 8DR engine. The proposals as made, arrived at after delegates had consulted with the Leningrad Shipbuilding Institute, centered around recommendations that the power rating of the engine be changed from 2,000 to 2,500-horsepower while, at the same time, effecting a reduction in fuel consumption of 5 percent. How? Unstated.

Meanwhile, the 18D engine, installed in the *Krasnoye Sormovo*-class diesel tug, has created its own problems for the Irtyish Steamship Line, operators of a considerable number of the class. The Omsk Ship Repair Yard, perhaps the largest enterprise of its kind in the Irtyish basin has, in the past, overhauled the engine in place on board ship. However, since most of the work, particularly that involved in a major overhaul, has been done during the winter period, results have been far from satisfactory. Machinists, working in freezing weather, in unheated engine rooms, have been prone to do shoddy work. The results of such work do not, of course, appear until spring, just when the tugs should be getting underway for the season. The consequences have been breakdowns, loss of operational time, and higher costs.

Last winter, for the first time, engines were pulled and the work was done in the shore shops. Plans for the operation were drawn during the preceding summer, at which time a special section for the work was set up in the yard's machine shop, to which 30 machinists were assigned. The area so set up is said to be adequate for repairing ten engines simultaneously. A new feature for the yard was the installation of a test stand so that the overhauled engines could be run-in under conditions approaching normal operations. Instead of drawing on spare parts, which appear to be in short supply, the machine shop, prior to the beginning of the overhaul period, began the manufacture of needed pistons, liners, rings and other parts.

By the beginning of the 1958 season the yard was able to announce that it had completed the major overhaul of 16 engines of the 18D type. The engines had been pulled, set onto special steel dollies and then drawn into the shop by tractors. Repair costs are said to have been reduced 7,000 rubles per engine. Inter-plant cooperation, whereby central manufacturing plants produce specific items for all consumers, broke down, however, and such items as pumps (fuel, lube, water), valves, bolts and nuts, were not delivered on time. Should this feature of the program show improvement, it is anticipated

that production during the coming winter repair season will be even better in Omsk.

Similar methods were employed for the first time in the Lenin Shipyard in Astrakhan, with results about the same as those reported for Omsk. The idea is slow in taking hold, despite the obviously better results, and the Director and Chief Engineer, Kamyishyev and Kolchanov respectively, of the Teplokhod Shipyard have been taken to task for dragging their feet in getting the method into practical use.

While old engines are having the bugs worked out, new engines appear. The latest to be announced is a model 6ChR-25/34 marine diesel engine. Rated at 300-horsepower, the engine is a six-cylinder unit, four-cycle, reversible, with a bore and stroke of 250 and 340 millimeters, respectively. The builder is the 25 October Machine Building Works in Pervomaysk, an organization subordinate to the Kherson Council for the National Economy. The engine is designed for use in small river and seagoing ships and can be used directly-coupled to the propeller shaft or in conjunction with a specially designed reduction gear and flexible coupling.

PROPELLER DIFFICULTIES

The Soviet maritime fleet suffers a considerable loss in operational time because propellers break down or require frequent replacement, the result of corrosion. The problem is particularly acute among ships fitted with low-carbon steel propellers, the usual period of service for which is from 2½ to 3 years. The motorship *Aleksandr Matrosov* (2,287 GRT), built in Finland in 1945, had two propellers replaced between 1949 and 1953. The tankers *Kreml* and *Iosif Stalin*, both built in 1932, similarly require propeller replacement every 2 to 3 years.

Yu. Zobachyev, Chief of the Corrosion Laboratory in the Central Scientific-Research Institute for the Maritime Fleet, admits the solution is to make the propellers from non-ferrous metals or from corrosion resistant steel, but he also points up a fact which has long been a source of complaint in the shipbuilding industry, there is a definite shortage of non-ferrous metals in the Soviet Union. Too, the technology in casting corrosion resistant steel is quite complicated and appears to pose problems not yet solved by Soviet engineers. As a result the Soviet Union operates many ships fitted with propellers made of low-carbon steel.

In an effort to solve the problem, the Institute has collaborated with the Kanonerskiy Shipyard, Central Planning-Design Bureau No. 1, and the Baltic Steamship Company in doing some experimental work in tipping propellers with strips of corrosion resistant steel 1.5-mm thick. The strips are first tack-welded to the blades and then welded along the edges. Mark TsL-9, Type EA-1B electrodes are used in the final welding process, after which the finishing and polishing, as well as balancing is done.

In December 1957 one such propeller was installed on the steamship *Volocha-Yevsk* (2,725 GRT), a cargo type built in Germany in 1943 and which usually operates in the Baltic for the Estonian Steamship Company. After several months of operation the propeller was inspected and appears to have withstood the ravages of time and corrosion well enough to suggest that regular use may be made of this approach to the problem in the future.

PLASTICS IN SHIPBUILDING

Recent decisions of the Communist Party in the Soviet Union have required a stepped-up tempo of development in the chemical industry, particularly that sector engaged in the development and manufacture of synthetic resins, plastics and glass wool. News of the use of plastics in shipbuilding comes from Leningrad.

M. Andriyevskiy, Chief of the Central Technical Design Bureau of the Ministry for the Maritime Fleet and M. Avrukh, Chief Designer, note that the imeni Kalinin Yard in Moscow is presently building lifeboats from plastics. Two types are in production. The one, a motor-less type, can be fitted with an outboard motor; the other is 50 feet long and with an inboard type engine can make 15-knots. Both were designed by the Bureau in collaboration with the Moscow Scientific-Research Institute for Plastics and the Technical Administration of the Ministry.

The material in use is a substance, the specific weight of which is one-fifth that of steel, and two and one-half times less than that of aluminum. Impervious to atmospheric conditions, the substance has retained its strength in a range of temperatures from minus 70 to plus 200 degrees Centigrade. It remains unaffected by gasoline, kerosene, oils, and salts which usually tend to corrode ordinary shipbuilding materials. It need not be painted.

Since the substance is readily molded, there is no need to take up valuable way space for construction purposes. Two men can handle a completed lifeboat, while the motorboat is said to be capable of carrying 40 percent more cargo than a steel boat of similar size. The plastic boat has a future as a cargo carrier for the very shallow rivers, since it can be produced cheaply, the man hours needed for construction being 40 percent less than for a similar steel or wooden boat. It is further pointed out that little space is needed to commence mass production, comparatively speaking. The imeni Kalinin Shipyard, for example, has the capability for 1959, using approximately the same machinery as at present, to build 1,100 lifeboats, 190 motorboats and 115 cargo motorboats for the small rivers, each of the latter capable of carrying 10-15 tons of cargo. A program of this scope would require 250 tons of polyethylene resin and 170 tons of glass wool. This past summer

was to have witnessed tests of all three types of boats described above under normal operating conditions.

Planning was also to have started for a number of other types of ships, such as dry cargo and tanker motorships of 50 and 100 tons capacity each, tugpushers of 30, 80 and 150-horsepower, barges for use with the pushers carrying 30 and 100 tons, passenger cutters for 50 and 100 persons, as well as small types of craft for individual use. Plastics are also recommended for expanded use throughout the shipbuilding industry both in the hull and superstructure.

There is an element of uncertainty about this ambitious program, however. It is concluded that, unfortunately, at the present time there is not enough polyethylene resin, glass wool, or other materials available for river transportation. Priorities appear to have directed available supplies into other fields and it will not be until the chemical industry has stepped up output considerably that river requirements can be met. However, it is urged that such requirements be made known to the industry by the Ministry and that requests be made to meet such requirements.

BOILERS

The average length of time between boiler cleanings for vessels operating for the Kama Steamship Line is said to have reached 2,400 hours for water-tube type boilers. Vessels equipped with this type boiler average 1.2 boiler cleanings per season, each cleaning taking 58 hours. Fire-tube type boilers, on the other hand, require an average 1.8 cleanings per season and the process takes 65 hours. Preliminary figures show that the line, competing for operating efficiency, saved better than 1,500,000 rubles on fuel during the 1957 season, or some 7,500 tons of oil, for example, at current prices.

Still on the subject of boilers, designers in the "Leninskaya Kuznitsa" Shipyard in Kiev have prepared the technical documentation for what is said to be the first Soviet marine type automatic feed water heater. The heater is to be used, evidently, on river vessels equipped with fire-tube boilers.

SHIPYARD NOTES

imeni A. A. Zhdanov

By the end of 1960 this Leningrad shipyard is to complete the building of a series of newly designed passenger motorships capable of carrying 260 persons. The new class was designed for use on the Black and Caspian Seas and will displace 3,000 tons. The first keel to be laid was that of *Kirgizstan*. Powered by two 1,000-horsepower diesels, the class will have air conditioning, while the superstructure and cabin bulkheads will be built of a light-weight aluminum-magnesium alloy.

Admiralty

Probably the builder of the atomic icebreaker, certainly builder of *Kazbek*-class tankers, this Leningrad shipyard has also built at least two such tankers for East Germany. *Leyna-1* and *Leyna-2* (named in honor of the largest chemical works in East Germany) are somewhat modified from the original *Kazbek* design, as are other, more recent versions for Soviet use. Such design changes include streamlining of the superstructure and stack, as well as the fitting of additional bulwarks to improve the ship's silhouette. Somewhat startling is the announcement that all metal furniture has been replaced by wood.

Leyna-1 was launched in December 1957, and was being completed in the spring of 1958 in the yard's basin. At that time *Leyna-2* was on the ways. The first of the two has been promised for delivery during 1958 and the prospective skipper, Captain H. Heimer, as well as the future chief engineer, E. Laue, were both in the building yard in the spring, the balance of the crew scheduled to arrive in midsummer.

The keel of a crab cannery for use in the Far East was scheduled for laying down sometime in 1958. Beyond the fact that the vessel will displace 14,000 tons, somewhat less than a *Kazbek*, and that it will be able to remain at sea for two months, nothing is known. The displacement approaches that of the icebreaker *Lenin* and may be the reason for some statement that a second such icebreaker is under construction.

Baltic Shipyard imeni S. Ordzhonikidzye

In addition to having under construction the first Soviet "super" tanker, which was scheduled for launching in October, 1958, the yard has begun the construction of a second series of refrigerator ships for use by the fishing industry. The new series is to be equipped with refrigerators of greater capacity, the major change reported.

Late in December the launching of *Krasnogvardeysk* was reported, at the same time that *Kamenogorsk* was placed in service with the fishing fleet. The launch took place with the vessel 74.7 percent technically ready. Main and auxiliary machinery were installed on the ways, as was shafting and steering equipment. A considerable part of the piping and electrical installation was also completed prior to launching. Dock trials were scheduled for the second quarter of 1958, at about the time that *Kramatorsk* was to slide down the ways.

Gomel

At one time known as the Gomel Shipyard, now the Gomel Repair-Operations Base, this Dnepr River shipyard has for some time been building 150-hp diesel tugs powered by the 3D6 engine. Scores of these tugs, or pushers, have, over the years, found their way to Lena, Yenisey, Amur, Pechero, Amudarya, Don and other rivers. Last year twelve tugpushers were launched and, in January 1958, the

yard started on the production of similar craft using water-jet engines. Future plans include the building of pontoons for floating cranes, a hydrofoil-design water taxi, as well as the *Raketa*-class hydrofoil craft.

Imeni Karl Marks

This Astrakhan shipyard has received an order to build eight ferryboats of an unspecified type and class for use in carrying tractors and other agricultural machinery for the collective farms. Five were said to have been completed in the spring, with delivery of the remainder sometime during the year.

Kherson

The Kherson Shipyard is credited with delivery of yet another *Kazbek*-class tanker, this one named *Uzhgorod*. By the latter part of April *Uzhgorod* was conducting underway trials, prior to assignment to the Black Sea Steamship Company.

The imeni Komintern Shipyard in Kherson, a river shipyard, has a production plan for 1958 which includes five 300-hp motorships and ten lake-type motorships of 150-hp, all of which will be turned over to the passenger fleet on the Dnepr River.

Krasnoyarsk

This shipyard is said to be the principal builder of shallow-draft vessels for the Ministry for the River Fleet. In the past ten years the yard has built over 680 various types of river craft ranging from 50 to 150 horsepower for the small rivers in the Ural region, the Far East, Siberia, and other regions. This yard was the first to build a water-jet propelled tug and today 55 vessels of this type are in use in Krasnoyarsk Kray, while 40 are in operation in Altay Kray, according to Director V. Kozmin.

Krasnoye Sormovo

Talk of a seagoing ferry for use on the Caspian appears to have been translated into action with a May report that this shipyard, using plans provided by the Central Design Bureau of the shipbuilding industry, had begun construction of such a vessel. When completed the ferry will be assigned to carry rail cars and passengers between Baku and Krasnovodsk across the Caspian Sea. Announced dimensions include:

Length overall	440 feet
Beam	60 feet
Height of side	20 feet
Draft, loaded	15 feet
Deadweight	2240 tons
Passenger capacity	300

The design calls for a single-decked, triple screw vessel displacing 6,000 tons at full load. Propulsion is to be provided by four 3D100 diesel engines developing 1,800-horsepower at 810-rpm, coupled to direct-current generators rated at 1,250-kilowatts. Total power of the diesel-electric installation will be 7,200-hp, sufficient to produce a speed of 16-knots. Three drive motors will be installed athwartships and will be shafted to four-bladed brass propellers. The outboard propellers will have a diameter of 8.4 feet, while the amidships propeller will be somewhat larger, 8.9 feet. The well deck will accommodate thirty 4-axle, 50-ton freight cars, or 16 all-metal passenger cars. The cars will be brought aboard from the stern over a special ramp from the shore on four sets of rails. The passenger spaces will be on the upper deck, over the car well. Two and four place cabins will care for 96 passengers, while the remainder will have the use of aircraft-type seats.

The crew has been set at 67 and the equipment to be installed will enable the vessel to operate in any weather. The present program calls for four such vessels and, when all four are in service, will carry annually an estimated 70,000 rail cars, or over 2 million tons of cargo. The annual saving by the use of loaded rail cars transported by ferry will reach an estimated 180 million rubles.

Leninskaya Kuznitsa

Located on the Dnepr River in the city of Kiev, is a trawler building yard which has also provided its plans for the Klaypeda Shipyard, on the Baltic. A group of the southern builders made a trip north to witness the launching of the first of the Klaypeda-built trawlers, *Komsomolets Litvyi*, in February. By late June the Kiev yard, in turn, was getting ready to launch a refrigerator motorship named *Ivan Khivrich*, the first vessel of the type built in the yard and one which will be used in lake-river traffic to carry perishables, particularly frozen meat and fish, as well as milk products, fruits and vegetables. Freon will be used. The twin-screw ship, with an all-welded hull, is fitted with two 300-horsepower engines and will make an estimated 10 knots. All auxiliary machinery is electrically propelled and main engine controls, in line with a current trend in Soviet shipbuilding, will be centralized in the wheelhouse, as well as in the engine room.

Two somewhat smaller versions of the above vessel, the motorships *Volkhov* and *Svir*, with two 150-horsepower diesels installed, and capable of carrying 600 metric tons, were delivered to the Dnepr Steamship Line in time for the river season. They are said to be equipped with the "latest" in navigation equipment.

Early in the year the yard built a diesel-electric dredge for the port of Haiphong, Viet Nam, then disassembled the unit and shipped individual sections to the port. The multi-bucket unit was then reassembled by a group of engineers and workers sent out from Kiev. The dredging equipment was mounted on a barge 145 feet in length, 31 feet in beam. The endless belt of 35 buckets can be dropped to a maximum depth of 8-meters. Each bucket has a capacity of 400-liters and movement of the belt can be adjusted within a range of from 7 to 17 buckets per minute. The 450-horsepower, four-cycle diesel drives a 300-kilowatt, 400-volt generator. It is understood that the unit is similar to previously

manufactured dredges provided for the Volga, Kama, and Don rivers in the Soviet Union.

Curiously enough, on 15 May Odessa port workers greeted the return to the Soviet Union of Suction Dredge 20 which, for the past two years, has been operating in the ports of Viet Nam. The dredge had left for Haiphong in January 1956 and, in the two years of work, is said to have removed 8½ million cubic meters of spoil. A Captain M. Ye. Kravchenko was in command of the dredge.

Maintaining its own production of trawlers, but this time for the north, the yard sent two units to Kaliningrad early in the year, with a third to follow later on.

M. Gorkiy

The scope of the *Zelenodolsk*-class program, a review of which appeared in the May 1958 issue of the JOURNAL, is suggested by the appearance in the Volga Tanker Steamship Line of seven such vessels for the 1958 season. A picture of *Pkhenyan* at a dock in Astrakhan appeared in late May. Yet another of the class bears the name *Pekin*, suggesting a series of Chinese city names will distinguish the class.

Moscow

On 30 June the Moscow Shipbuilding-Ship Repair Yard, builder of "hundreds" of *Moskvich*-class passenger motorships, completed the construction of three of the class for shipment to the Yana and Kolyima rivers, in Siberia. The two assigned to the Yana will operate over a distance of 450 miles, between the ports of Ust and Verkhoyansk. Because of the distance, as well as because of the climate, the vessels have been fitted with a small galley, bathing facilities, improved heating systems, and powerful radio equipment. All three units were to be sent to Arkhangelsk, there to be loaded aboard sea-going lighters and towed to their final destinations via the Northern Sea Route.

imeni I. I. Nosenko

Nikolayev, in addition to its other activities, was to contribute to an augmentation program for the fishing fleets of the Baltic in 1958 by providing six large trawlers, powered by 2,000-horsepower engines. One of these is *Maya-Kovskiy*, the holds of which have been equipped with canning, fish handling, oil extraction and other shops, as well as with a large freezer installatiton.

Late in February it was announced that a tanker bearing the name *Nikolayev-Skiy Pioner* was under construction, apparently the latest of the *Kazbek*-class to be delivered by this yard.

In addition the latest of the series of ships with names ending with the letters GES, an abbreviation transliterated from the Russian words for hydroelectric station, went into service early in the year. The full name—*Tsimlyanskges*, a diesel-electric. An early ship of the type is shown in Figure 4.

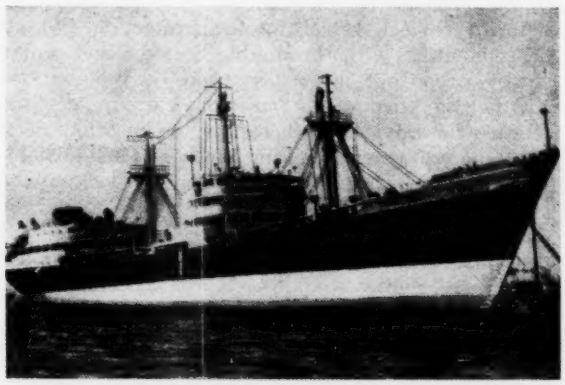


Figure 4. The diesel-electric *Dneproges*, 5494 GRT, built in the I. I. Nosenko Shipyard, Nikolayev, in 1956. Six of the class are known to be in operation at the present time.

Riga

The Riga Ship Repair Yard outfitted the motorship *Ivan Moskvin* with an unspecified type of radar, but presumably the Neptun, in addition to other navigational equipment when the vessel arrived from its building yard in Belgium early in the spring.

Samus

This shipyard on the Ob River, designated as a Repair-Operations Base, has joined the ranks of shipyards in the Soviet Union turning out the *Moskvich*-class passenger vessel. The base is also making plans to build a new type of shallow-draft vessel rated at 300-horsepower and using a water jet propeller. Two of the latter are supposed to have gone into service this season on Ob River tributaries as passenger carriers.

imeni Stalin

Usually employed in the repair of tugs used by the Volga Tanker Steamship Line, the Astrakhan yard is now engaged in a program of building steel dry-cargo barges, motorboats and self-propelled ferry boats. Mention is made of the building of Ferry Boat Number 3, suggesting that this phase of the program may be well along. No mention is made of the use to which these ferries may be put, but it is known that such transportation is highly desirable on the Caspian Sea, at least in the Soviet planners' view.

Stalingrad

The Stalingrad Shipyard has, in the past, built 2,000-ton capacity cargo motorships, lake-type barges and other craft. In late May it was announced that the first river tanker built in this yard had been launched. It will be able to carry 3,300-tons of fuel, including viscous types requiring heating prior to discharge. Tank cleaning equipment will be installed, an interesting innovation, with which it is said tanks previously filled with black oil will be cleansed so as to load gasoline a few

hours later. The vessel, for fire protection, is built on twin hulls, with the double bottoms and piping so arranged that inert gases can be released for smothering purposes in case of fire. An unusual feature for Soviet river craft is the installation of pumping equipment, making the vessel self-sustaining. The two cargo pumps to be installed will be able to discharge 190,000 gallons per hour. The tanker, as may have been suspected, is built on the *Bolshaya Volga* design, with a hull suitable for use on the reservoirs as well as in ice. The engine installation will develop 1,000-horsepower for a speed of 9 knots. A 17 man crew will be carried, as opposed to a 23 man crew for the dry cargo carrier. It is expected that one additional tanker will be programmed in 1958 and two in 1959, with the total program in the vicinity of ten. Initially the tankers will operate on the run from Astrakhan to Ryibinsk to Perm. For the future, plans call for a 5,000-ton capacity tanker for the rivers.

Continuing its production of the 2,000-ton capacity cargo carriers, the yard, at mid-year launched *Aktyubinsk* and *Astrakhan*—not for the Volga, but for the Dnepr River. Transfer was via the Volga-Don Canal. Two additional units in the same series, *Akmolinsk* and *Armavir* were to have followed.

Tyumen

The Tyumen Shipyard, dating back to the beginning of the Soviet regime as a major producer for the rivers of the Ob-Irtyish basin, is currently building five types of river ships. One of the types is a three-decked passenger motorship with 308 cabin spaces. The first was to be launched during the summer. A second type is a self-propelled automobile ferry for the Bukhtarminsk Reservoir. Very nearly 200 feet in length, 45 feet in beam, it will be able to carry twenty loaded trucks of the ZIL-150 type on its deck. Accommodations for 200 passengers are provided on this double-ender type ferry.

Yet a third type is the self-dumping timber barge described in the May 1958 JOURNAL. The 1958 production plan for this type called for twenty units. In mid-March six of them were already being assembled on the ways and it was anticipated at that time that by the opening of the timber-carrying season ten of the barges would be in operation on the lower reaches of the Irtyish River, with an additional ten coming into service by the end of the season. A fourth type is a passenger motorship similar to the *Leninskiy Komsomol,* which operates on the run from Omsk to Salekhard. Tanker barges appear to conclude the current program.

Future production involves tankers, shallow-draft motorships, refrigerators, and continued production of ferry boats.

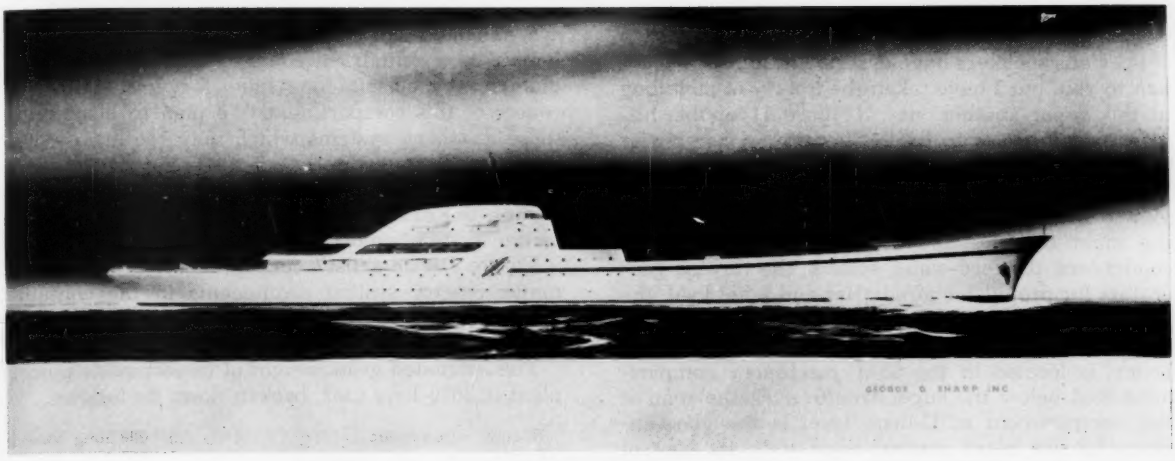
imeni Vano Sturua

In addition to the two steamship companies on the Caspian Sea, the Caspian Steamship and Caspian Roadstead, a third, known as the Caspian Petroleum Fleet, operates under the jurisdiction of the petroleum industry. This latter organization has received, from the imeni Vano Sturua Shipyard in Baku, a new type of seagoing tug. As distinguished from previously built roadstead-type tugs, this small unit will be able to operate in any part of the Caspian in any kind of weather. It will be rated at 1,200-horsepower and, during trials held in March, demonstrated good seakeeping qualities. The type was designed for doing geophysical work in distant areas of the sea, for rescue and salvage work in the areas where oil well drilling is done into the sea bottom, and for towing work. One such may be named *Transportnik* and a series will be built not only for the Fleet designated, but for the two companies under the Ministry for the Maritime Fleet as well.

A rather unusual announcement from this yard occurred when it was stated that two barges for use on both the Caspian and Azov seas had been launched. For use in bunkering operations, they are all steel, with some degree of welding having been used. The unusual feature is that use of the barges on the Azov Sea will require transfer out of the Caspian via the Volga-Don Canal, although, since size was not indicated, transfer by rail cannot be ruled out.

Zaporozhye

Yet another ferry boat type came into view with the report in mid-Spring that the Zaporozhye Shipbuilding-Ship Repair Yard had completed its first self-propelled vessel of this type. First of its type on the Dnepr River, and called SP-1, it is currently in use between Kherson and Golaya Pristan. Designed by the steamship line's Design Bureau, it crosses the Kakhovka Reservoir with tractors and trucks, as well as passengers. Its two 150-horsepower engines produce a speed of 8.5 knots, with a load of 600-tons. The yard is scheduled to lay down two additional vessels of the same type. The formal name of the vessel now in service is *Severodvinsk.*



N.S. Savannah

THE POWER PLANT FOR THE FIRST NUCLEAR MERCHANT SHIP N. S. SAVANNAH

ACKNOWLEDGEMENT

This paper was presented at the 1958 Nuclear Merchant Ship Symposium which was held in Washington, D.C. on 21 August 1958. It was prepared and presented by Mr. John W. Landis who is Assistant Manager of the Atomic Energy Division of the Babcock and Wilcox Company. This company holds the prime contract for the propelling machinery for the ***Savannah.***

The 1958 ***Savannah*** *story differs in many ways from the earlier one told by her predecessor, the first steam ship to cross the Atlantic. The basic difference is not in pioneering which is not too dissimilar. It is in the tremendous growth of the engineering profession in the intervening years. This article is a simple story of the practice of engineering in a professional and sophisticated way. Such an article could not have been written about the earlier* ***Savannah.*** *She was a craft product. Today's* ***Savannah*** *is a sound engineering, soundly engineered, project approaching completion. The paper is published with the permission of the Babcock and Wilcox Company.*

FINAL design and fabrication of the power plant for the first nuclear merchant ship, the N.S. *Savannah,* is proceeding in accordance with the accelerated schedules laid down at the outset of the contract. The Babcock & Wilcox Company and its primary subcontractor, DeLaval Steam Turbine Company, are now engaged in manufacture of the major components of the reactor system and the propulsion system, respectively. Looking back on some of the troubles we have been through to date, we hesitate to venture a guess as to what the future has in store. With a large portion of our present and potential market staring us in the face, however, you may rest assured that we shall leave no turnable stone unturned to complete this job on time and in accordance with specifications.

Of course we cannot predict with absolute certainty at this stage what the performance of the

Nuclear Merchant Ship Reactor will be. As a minimum, we shall meet our guarantees, which have of course been set reasonably low in many respects. Taking an optimistic viewpoint, on the other hand, the first-core cruising range could be as much as 500,000 miles.

PLANT LOCATION, LAYOUT AND PERFORMANCE DATA

Previous speakers have described the N.S. *Savannah* to you, but I have taken the liberty of including in this paper another view (Figure 1) of this historic ship in order to fix the location of her power plant more definitely.

Figure 1 is an elevation of the ship and a plan view of D deck. The propulsion system, comprising the main turbines and reduction gear, the main condensers, the feed-water system, the turbine generators for propulsive auxiliaries and hotel load, the low-pressure steam generator, and the emergency diesel generators, take-home motor and package boiler, is located in the 55-ft. machinery compartment just below the superstructure. At the rear of this compartment at D-deck level is the glass-enclosed power plant control room with its control console and various instrument panels. A visitors' gallery runs across the front of the control room.

The reactor system is located in a special 67.5-ft. compartment just ahead of the machinery space—about amidships. The major portion of the reactor system, including the reactor itself, the primary coolant loops, the pressurizer, the steam generators, the primary circulating pumps, most of the auxiliary systems, and an air-conditioning system, will be housed in a cylindrical containment vessel 35 ft. in diameter by 50.5 ft. long lying fore-and-aft in the center of this compartment. We plan to place only those auxiliary systems which may require special maintenance outside the containment vessel—the primary water demineralizers and filters, the charge pumps, the drain tanks and the gaseous waste collection system, among others.

Figure 2 is an artist's concept of the layout of the major reactor system components in the containment vessel. The principal power plant performance data are given in Table 1.

The estimated gross weight of the complete power plant is 3650 long tons, broken down as follows:

Reactor equipment	600 long tons
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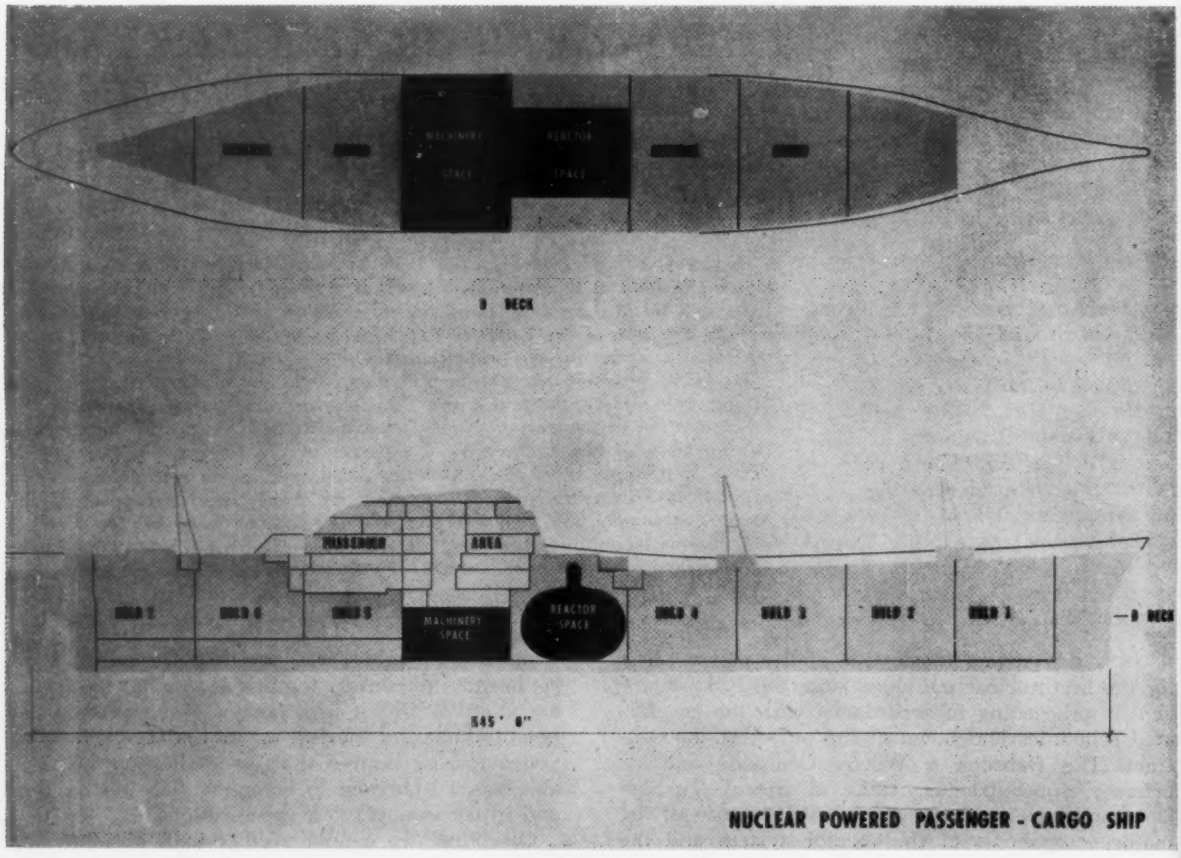

Figure 1.

TABLE I—*Principal Plant Performance Data*

OPERATING CONDITION	Normal Power	Maximum Power
Shaft Power—SHP	20,000	22,000
Propeller Speed—RPM	107	110
Turbine Inlet Pressure—psia	470	450
Main Condenser Pressure—"Hg abs.	1.5	1.5
Boiler Drum Pressure—psia	490	474
Feedwater Temperature—°F	347	343
Total Electrical Load—kw	1,900	1,900
STEAM CONSUMPTION—LB/HR		
a. Main Turbines	186,610	205,100
b. Feed Pump Turbine	24,750	25,500
c. Turbogenerators	28,280	28,280
d. Low Pressure Steam Generator	705	705
e. Air Ejectors, Sep'rs and Losses	1,855	1,875
Total Generation Required	242,200	261,460
HEAT TRANSPORT—MW		
a. Steam Generators	62.8	68.3
b. Purification System	1.2	1.2
c. Primary Pumps	—0.7	—0.7
d. Heat Losses	0.2	0.2
Reactor	63.5	69.0
ENGINEERING CHARACTERISTICS		
Primary Loop Pressure—psia	1750	1750
Primary Loop Pressure Drop—psi	69.5	69.5
Primary Loop Flow—lb/hr	8,000,000	8,000,000
Primary Loop Temperature Rise—°F	22.8	24.8
Primary Loop Mean Temperature—°F	508	508
Blowdown—lb/hr	2,400	2,600

REACTOR SYSTEM

Reactor Proper (Refer to Table II)

The primary source of power for the N.S. *Savannah* is a pressurized water reactor operating at 1750 psia and fueled with uranium oxide of about 4% enrichment clad in stainless steel rods. The reactor design has been influenced largely by the objective of a long core reactivity lifetime. The design target is approximately 52,200 MWD or 1230 days of operation under normal conditions.

The active core of the reactor is approximately a right circular cylinder, 62 in. equivalent diameter by 66 in. high. The core is made up of 32 fuel elements, each about 8.5 in. square and approximately 85 in. in overall length including end fittings. The fuel elements are confined within a semi-permanent "egg-crate" type lattice which provides the equivalent of a pressure can around each element. The elements are thus positioned in a square array on a 9.7 in. pitch, as illustrated in Figure 3. The present design of the "egg-crate" lattice calls for a 0.109 in. stainless steel wall in the second-pass region and a 0.094 in. stainless steel wall in the third-pass region to withstand the pressure differentials that arise as a result of the multi-pass flow pattern. Reactivity control is provided by 21 cruciform control rods, 8.0 in. tip to tip. The control rods are .375 in. thick, of a composite design, with a 3/16 in. plate of boronated stainless steel sandwiched between two 3/32 in. ordinary stainless steel plates.

The reactor internals are also illustrated in Figure 3. The entire core and the thermal shields are supported by the core support skirt, which is mounted on the vessel wall near the upper flow nozzles. The core is completely enclosed by the inner thermal shield and lower flow baffle.

The "egg-crate" lattice containing the fuel elements is confined between a lower and an upper grid plate which are approximately 92 in. apart. The core and internals are held down against the upward thrust of the coolant flow and against downward hull accelerations by the upper baffle hold-down spring. The downward force of this spring is transmitted to 18 equally spaced vertical struts which bear on the upper grid plate. The internals are designed to prevent excessive motion of the core under the worst conditions of roll, pitch, and heave of the ship, as specified by the Maritime Administration. They will also hold the core in place if the ship capsizes.

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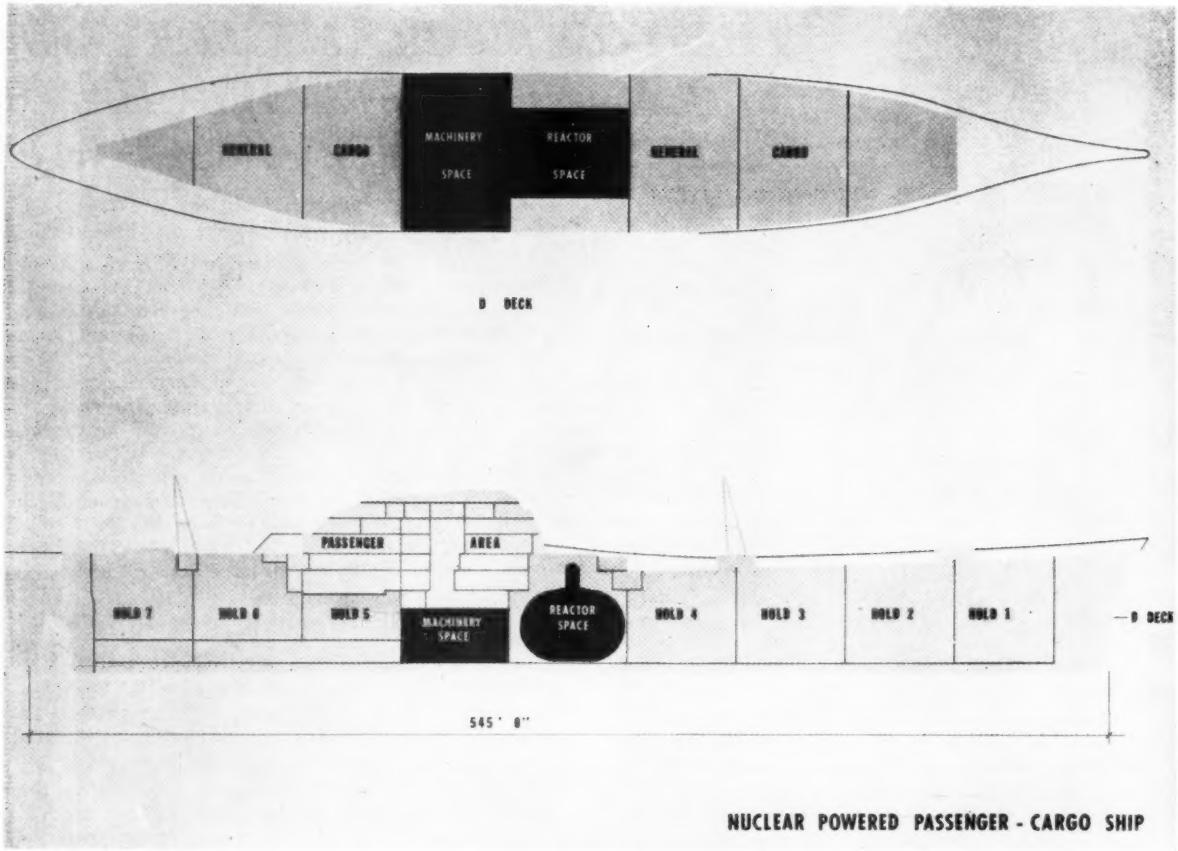

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TABLE II—*Reactor Design and Performance Characteristics Thermal, Hydraulic, and Nuclear*

CORE DIMENSIONS

Core over-all length		90.24
Active fuel length		66 in.
Equivalent core diameter		62.06 in.

FUEL ELEMENT DATA

Fuel		UO_2
Cladding		Boron stainless steel
Ferrules		Stainless steel
Springs		Inconel-X
Can		Stainless steel
Straps		Stainless steel
Fuel pellet diameter		0.4245 in. (nominal)
Cladding outside diameter		0.500 in. (nominal)
Cladding thickness		0.035 in. (nominal)
Fuel rod spacing (square lattice)		0.663 in. (nominal)
Fuel rods per element		164
Ferrule outside diameter		0.4375 in. (nominal)
Ferrule inside diameter		0.3975 in. (nominal)
Length of ferrule		1.0 in.
Fuel elements in core		32
Fuel element can thickness	(second pass	0.109 in.
	(Third pass	0.094 in.
Heat transfer area		3778 ft^2

THERMAL AND HYDRAULIC DATA

Reactor design power	74 MW
Reactor maximum operating power	69 MW
Reactor normal operating power	63.5 MW
Design pressure	2000 psia
Operating pressure	1750 psia
Total reactor flow rate	8.0×10^6 lb/hr
Velocity (average, second pass)	9.29 ft/sec
Velocity (average, third pass)	8.40 ft/sec
% leakage flow (second pass)	5%
% leakage flow (third pass)	10%
% total leakage flow	15%
Number of passes within reactor vessel	3
Bulk coolant inlet temperature at design power	494.7 °F
Bulk coolant outlet temperature at design power	521.3 °F
Average coolant temperature at design power (second pass)	499 °F
Average coolant temperature at design power (third pass)	512 °F
Core average coolant temperature at design power	508 °F

	Nominal* Channel (Third Pass)	Hot Channel With Flux Peaking (Third Pass)
Maximum coolant temperature, °F	536	543
Maximum surface temperature, °F	610	624
Maximum fuel temperature, °F	3660	4434
Power to start local boiling, percent of design power	113%	91%
Power to start net boiling	285%	233%

NUCLEAR DATA

Metal/H_2O ratio	0.76
Volume fractions	
Water	.5676
Control rods	.0397
Helium gap	.0064
Fuel	.2455
Stainless Steel	.1408
Typical Inventory and Fuel Burnup Data for 52,000 MWD Core Life	
Initial enrichment (wt %)	4.7%
Initial U-235 loading	331.4 Kg

Initial U-238 loading	6734 Kg
Average burnup	7360 MWD/Ton
Final enrichment (wt %)	3.9%
Final U-235 loading	272.9 Kg
U-235 consumed	58.4 Kg
Final Pu-239 loading	17.4 Kg
Final total Pu loading	19.6 Kg
Percent metal atom burnup	1.2%
Average thermal neutron flux	7.5×10^{12}

* The nominal channel is assumed to be located in the region of maximum flux but does not include the effect of manufacturing tolerances.

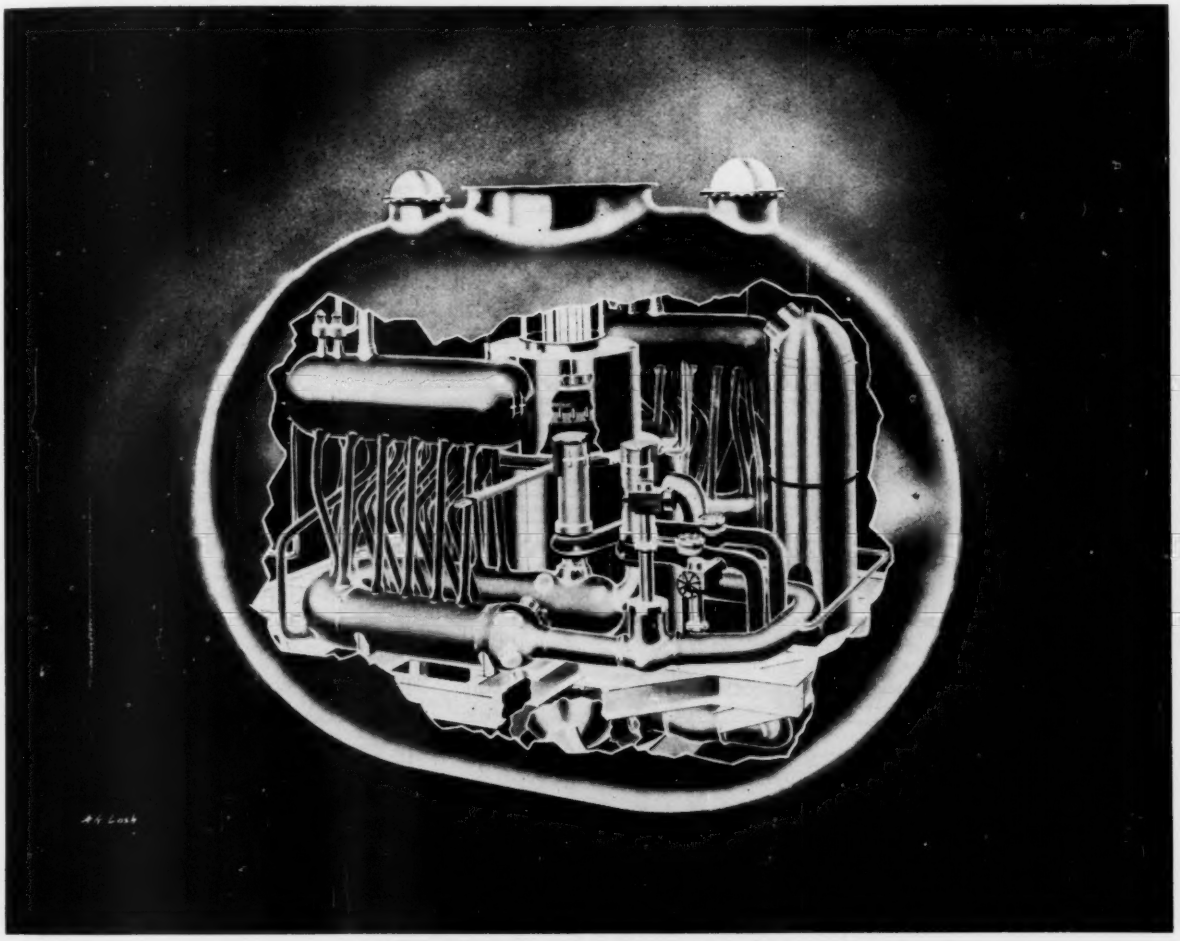


Figure 2. Reactor & Heat Transport System.

provides a manifold in which the primary flow is turned downward into the 16 outer fuel elements of the second pass. Flow tubes below the lower grid plate serve as an extension of the "egg-crate" lattice, directing the down-flowing coolant from the 16 outer elements into the lower plenum. These flow tubes also serve to prevent cross-flow and the accompanying lateral thrust against the control rod extensions.

At least 85 percent of the total primary flow from the lower plenum enters the 16 inner flow tubes that serve as an hydraulic entrance section for the fuel elements of the third pass. Approximately 9 percent of the total flow is orificed into the control rod channels. The remaining 6 percent is allocated to miscellaneous leakage, including a small quantity bled into the reflector region. The 15 percent "by-pass" flow reaches the upper plenum through the upper baffle plate.

Flow tubes above the core carry the flow from the inner elements directly to the upper plenum of the vessel, and at the same time prevent cross-flow and lateral thrust against the control rods in the "out" position. The various flows mix in the upper plenum and turn downward to the exit nozzles.

The fuel elements, as shown in Figure 4, are made up of four separate bundles of full-length fuel rods. Each rod consists of a 0.500 in. OD tube of 0.035 in. wall thickness filled with uranium oxide pellets of 0.4245 in. nominal diameter, pressed and sintered to 91 percent of theoretical density. The tube material is boronated type 304 stainless steel. The gap between the fuel pellets and the tube wall contains helium which is introduced during the end-plug welding process. Each of the four bundles comprising a fuel element contains 41 rods in a square array. Center-to-center spacing of the rods is .663 in., giving an over-all metal-to-water ratio of 0.76. The rod spacing is maintained by small tubular stainless steel ferrules brazed in place approximately every 8 in. along the length of the element.

Also shown in Figure 3 is the pressure vessel for the reactor. It is 26.5 ft. high and 98 in. inside diameter, is made of 6 in. carbon steel plate clad on the

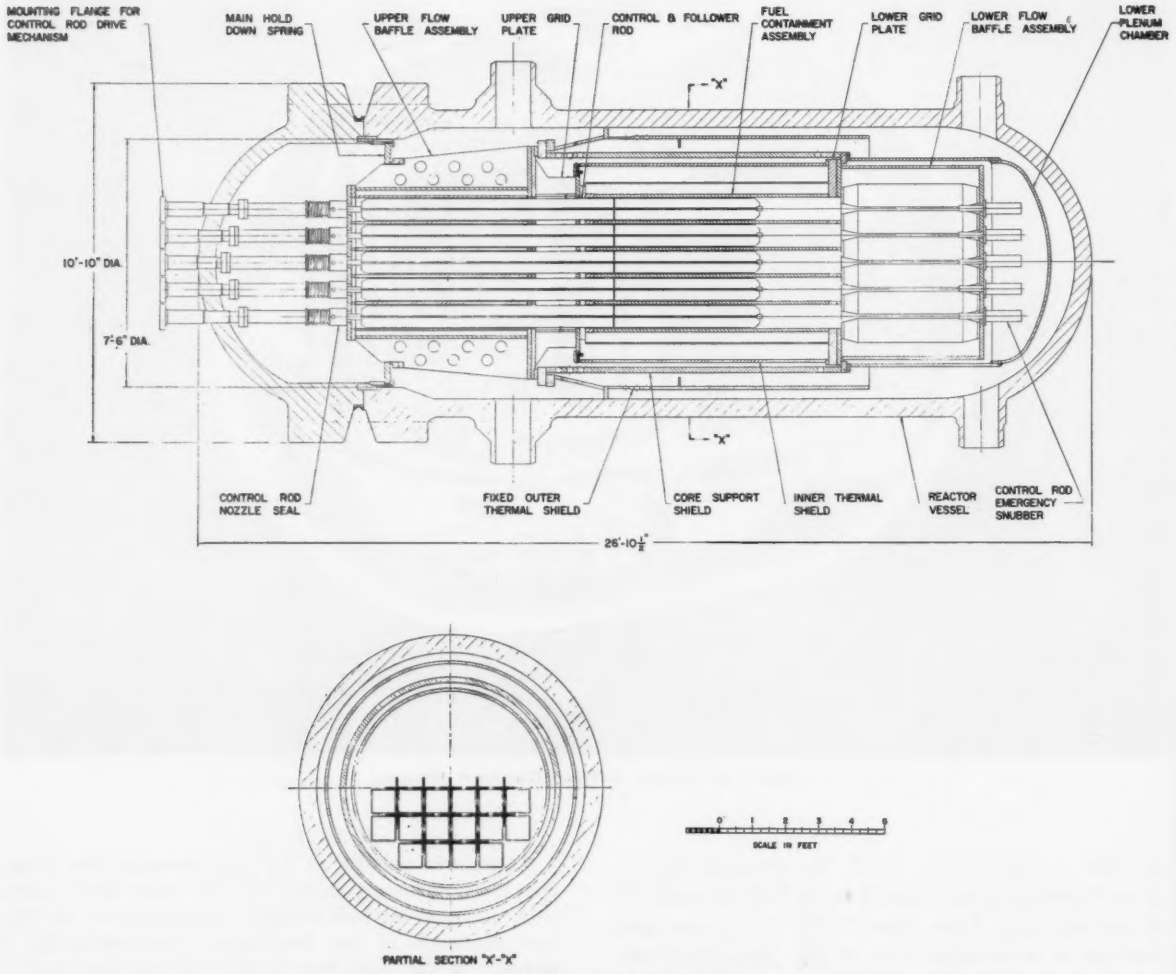


Figure 3. Reactor Vessel & Internals.

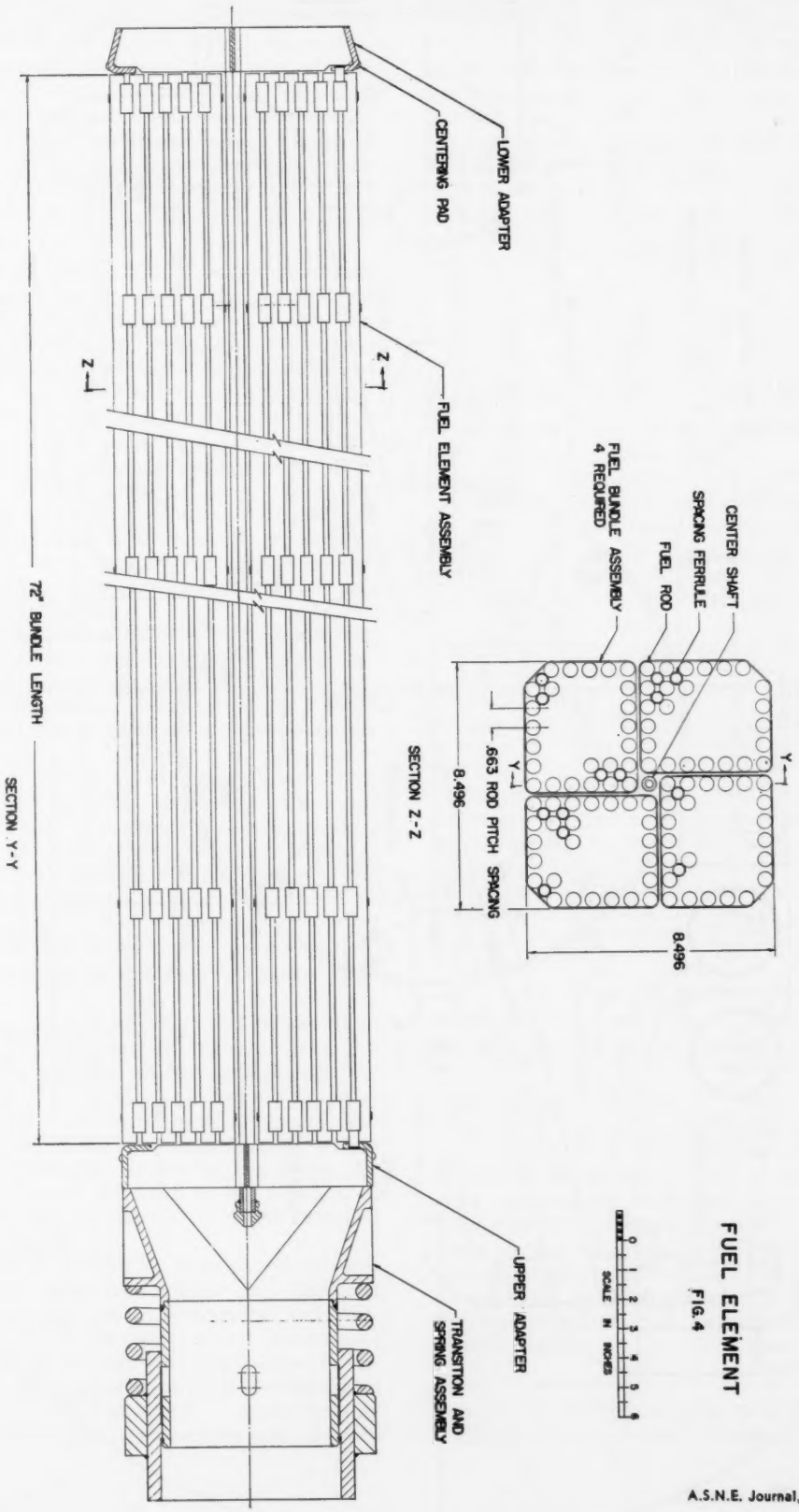


Figure 4.

FUEL ELEMENT
FIG. 4

SURGE LINE
4'-8¼"
17'-8¼"
STEAM DRUM
HEAT EXCHANGER
METAL SHIELDING
PRESSURIZER
REACTOR INLET
GATE VALVE
REACTOR OUTLET
CHECK VALVES
GATE VALVE
PUMPS
15'-5¼" O.D. SHIELD TANK
9'-2½"
9'-2½"
5'-0"
5'-0"
6'-3"
REACTOR
SHIELD WATER TANK
PUMPS
3'-9"
REACTOR OUTLET
GATE VALVE
CHECK VALVES
GATE VALVE
REACTOR INLET
INSTRUMENT THIMBLES
HEAT EXCHANGER
STEAM DRUM

inside with type 304 stainless steel. Its design pressure is 2000 psi and design temperature 650°F. The upper head, which carries the control rod drives, may be removed for refueling. Closure is effected by means of 48 five-inch studs with a double gasket and a seal weld between the flanges. The closure opening is 90 in.—sufficiently large for removal of all core components which may deteriorate or be damaged during the life of the plant.

Arrangement and Functional Operation of Reactor System

Figure 5 shows a reasonably accurate plan view of the arrangement of the major components of the reactor system within the containment vessel.

The reactor vessel is set vertically in a central position with the steam generators on each side and the primary piping laid out in left- and right-handed symmetry. This arrangement provides a minimum volume in the primary loops and minimum volume for the containment without exceeding permissible flexing stresses in the piping.

Figure 6 is a flow sheet showing the functional operation of the main coolant loops, the purification system, the make-up system, the pressurizing system, and a portion of the intermediate cooling system.

The primary water is circulated through the core by four pumps (two in each loop) and then to the tube side of the steam generators, giving up part of its heat to produce steam on the secondary side.

A continuous flow of 8,000,000 lb/hr and a constant mean temperature of the water through the reactor of 508°F is maintained with all four pumps in operation at all sea loads. Thus, at the maximum power of 22,000 SHP the reactor will be generating approximately 69.0 MW of heat, the water temperature rise through the reactor will be 24.8°F and the water will arrive at the steam generators at 520.4°F. Steam is delivered to the turbines at a pressure which rises with decreasing loads from a minimum of 450 psia at maximum power to 730 psia at zero power.

The primary loop is pressurized to 1750 psi by an electrically heated pressurizer vessel in which a steam space is maintained in equilibrium with the water by alternate use of electrical heaters or spray-cooling as the transients of the system demand. The water level in the pressurizer is maintained within the allowable fluctuation limits by regulating the rate of primary water let down to the purification system and varying the amount of water returned by the charge pumps.

The purification system is designed to remove continuously all particulate matter and dissolved solids in the primary water, both active and inactive—chiefly corrosion products. It will also remove fission products should any fuel rod failures occur.

During normal operation, water is withdrawn from the primary loop at a rate of approximately 20 gpm. It is reduced in temperature and pressure to about 110°F and 50 psig by means of a water-cooled heat exchanger, a control valve, and a block orifice and then is forced through the demineralizers and filters to the buffer seal surge tank. Gaseous fission products, if any, will accumulate here. The purified flow is then returned to the primary loop through a charge pump. Ordinarily the major portion of this flow will be returned through the buffer seals on the control rod drives, but a direct charging line with regulating valve is also provided capable of handling up to a maximum of 60 gpm. This is to permit a high rate of purification after reactor shutdown, if necessary.

There are three demineralizers, each having a minimum estimated life of 50 days so that if they are used in sequence the plant may be operated for at least 150 days before complete replacement is required. Please note that the lifetime is a minimum estimate. We fully expect a lifetime two or three times as great in actual practice.

Makeup water is added to the reactor system as required by means of makeup pumps which take suction from the deionized water storage tanks. Makeup water for the steam system is drawn from the ship's evaporator through a demineralizer into the main condenser. The same limits of purity are maintained in both primary and secondary water to avoid chloride stress corrosion in the boilers.

A hydrogen addition system (not shown) is provided to maintain a minimum concentration of 20 cc/liter of dissolved hydrogen in the primary loop to scavenge oxygen liberated by irradiation in the core.

The intermediate cooling system furnishes fresh water at 95°F to cool such components as the primary circulating pumps, the primary shield tank, the containment air conditioning unit, the letdown coolers and the gaseous waste collection system. This water is cooled in turn in an intermediate heat exchanger by sea water from a special sea chest for this purpose.

The containment air conditioning system continuously circulates and cools all of the air in the containment vessel, maintaining an average ambient temperature of 130°F and a humidity of 72 percent.

Reactor System Control

In steady operation the reactor system will respond automatically to the slight variations in steam demand at the turbine throttle by virtue of its negative temperature coefficient. No manual control will be necessary. During normal maneuvers, automatic control rod operation will cause the reactor to respond without serious lag or deviation in mean temperature. A graphic display of the operation of the reactor system will be provided on the main control panel.

Steam Generators

As shown in Figure 7, the two steam generators each consist of a U-shell, U-tube boiler section ap-

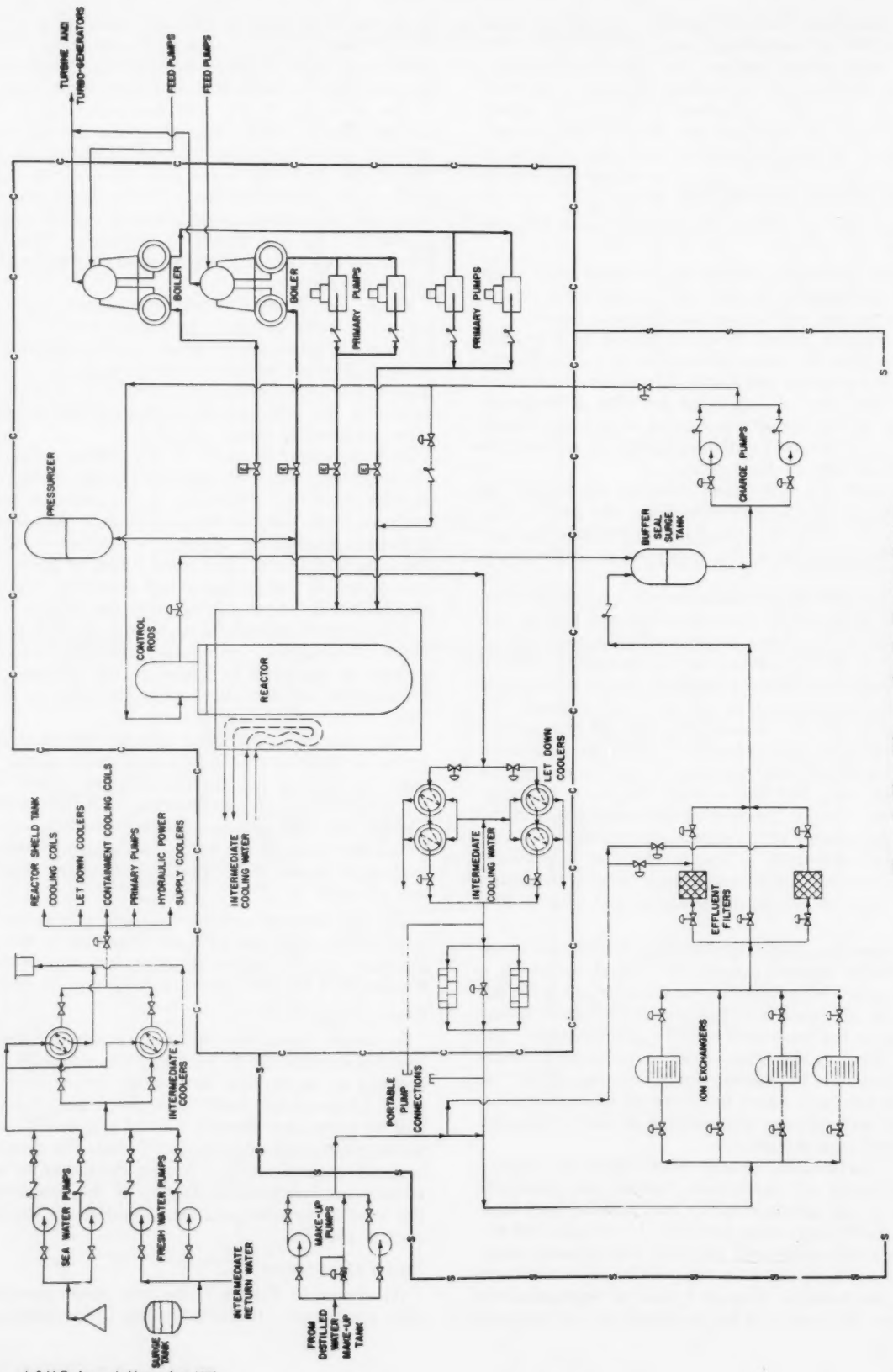


Figure 6. Flow Diagram for the Reactor System.

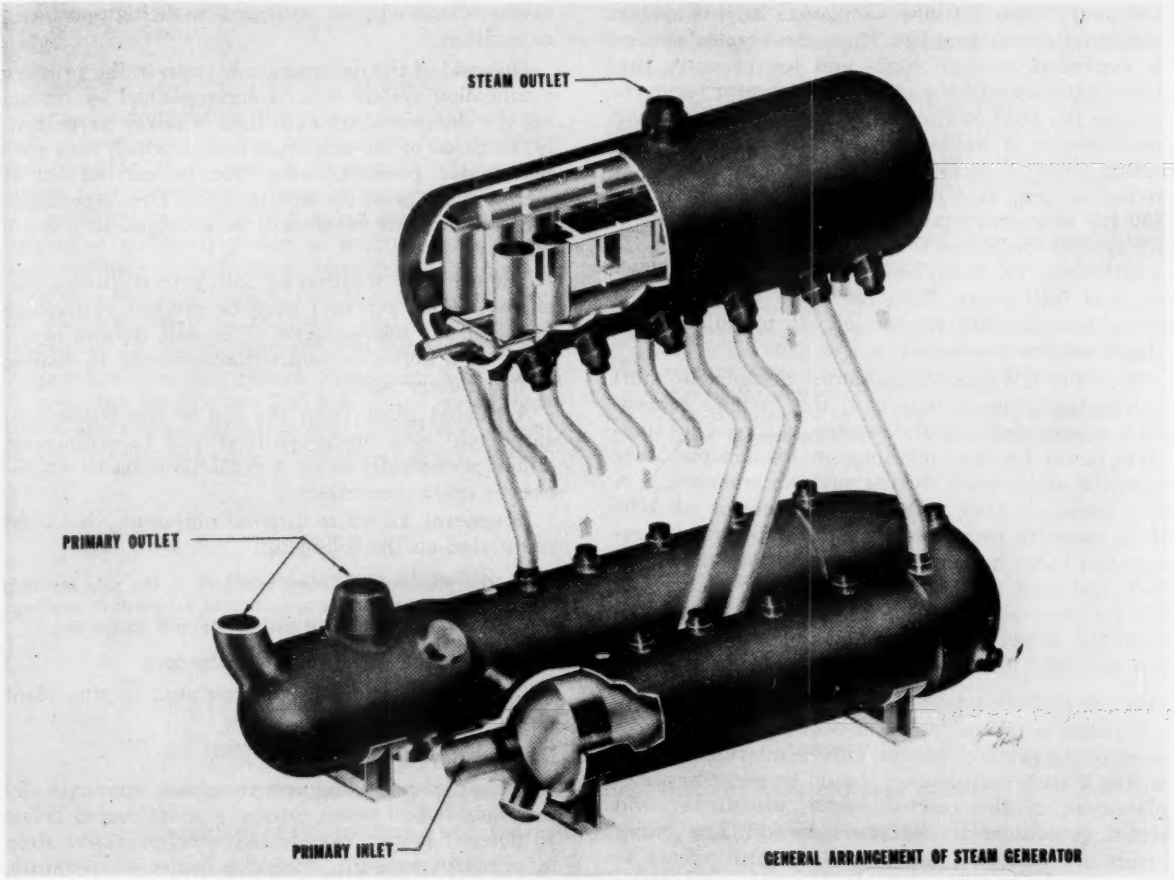


Figure 7.

proximately 20 feet long overall and contain about 800 stainless steel tubes ¾ inches O.D. These boilers are connected by the necessary risers and downcomers to steam drums overhead which are equipped with cyclone separators and scrubbers. The design pressure of the secondary side is 800 psi and the units are designed to transfer 74 MW of heat from the primary system at a steam pressure of 455 psia. They are equipped with conventional feedwater control, blowdown and chemical feed systems. The 8 in. steam pipe from each carries a quick-closing pneumatic valve as well as the usual stop-check valve.

Primary Circulating Pumps and Piping

The four primary circulating pumps are of the canned rotor type, each having half and full speed windings. They are designed to be all in continuous use during normal operation of the plant. The main design characteristics are as follows:

Power	250 KW
Flow	500 gpm at 495°F
Head	70 psi

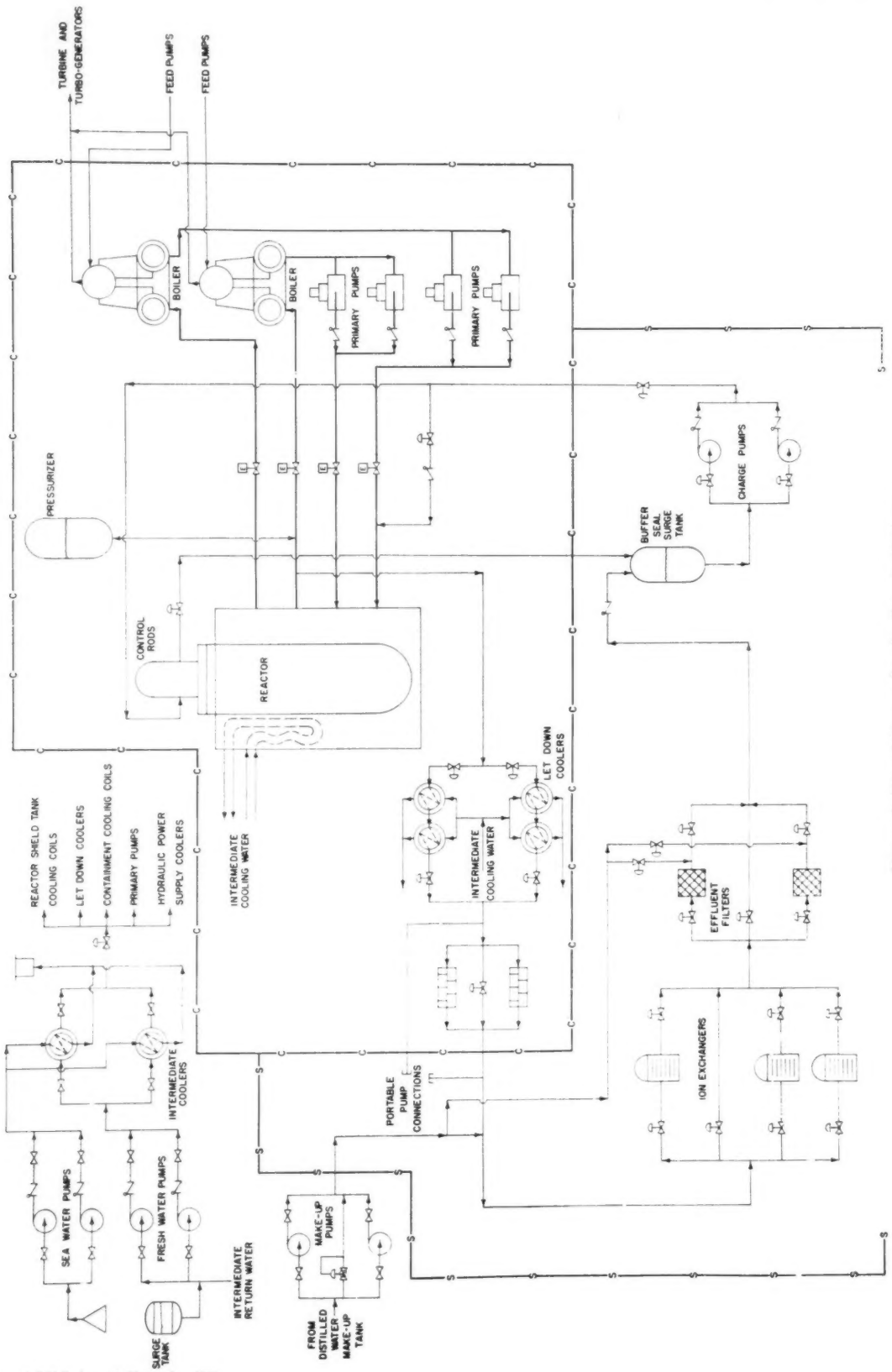
The main primary piping is 12-9/16 in. I.D. and the branch piping between the boiler outlets and the Y-fittings, containing the pumps, is 8-9/16 in. I.D. All primary piping is made from type 304 stainless steel hollow forgings with a design pressure of 2000 psi. Each primary loop also contains two main gate valves by which its boiler and pumps may be isolated from the reactor. A check valve is provided downstream of each pump.

Control Rod Drives

The 21 control rod drives mounted vertically on the head of the reactor are of the electro-mechanical type for reactor control but are provided with hydraulic cylinders for reactor scram. Each drive line is buffer-sealed where it passes into the head. Each seal is charged with purified water at 1800 psi. In the event of a seal failure, the seal is isolated with a backseating valve. Each control rod drive assembly may be individually disconnected from the control rod below and removed from the head.

PROPULSION SYSTEM

As stated previously, the major components of the propulsion system are being manufactured by the



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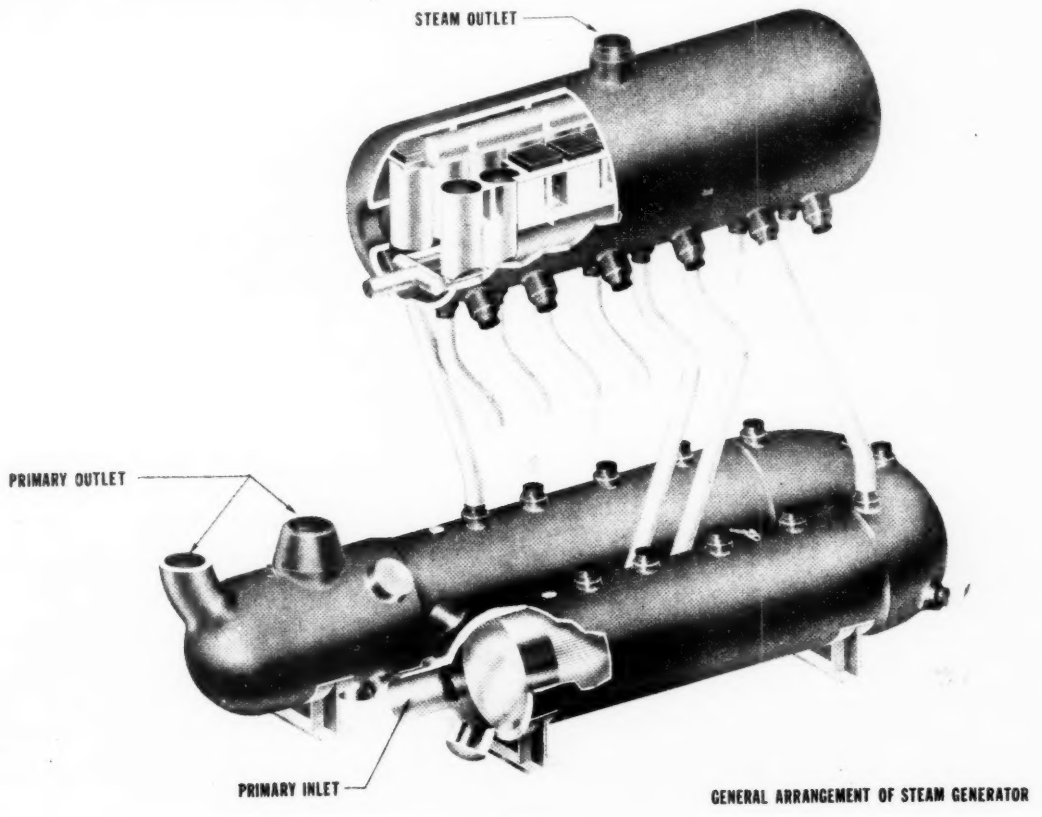

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DeLaval Steam Turbine Company. In this system saturated steam supplied from the reactor system is expanded through high- and low-pressure turbines between which a moisture separator serves to reduce the final exit moisture and losses. The main condenser is of welded tube-sheet construction and designed for 1.5 in. Hg at 75°F sea water. The double reduction gear incorporates means for coupling the 750 HP emergency take-home motor. The feed-water system consists of three feed-water heaters with a turbine-driven main feed pump and a motor-driven port feed pump. Two 1500 KW steam turbine generators provide power for all propulsive and ship's service auxiliaries.

Two 750 KW diesel generators, designed to start automatically upon failure of the turbine generators or upon scram of the reactor, will provide sufficient power for the take-home motor and the other essential ship's loads during such an emergency. A low pressure (110 psi) steam generator of 7500 lb/hr capacity provides steam for ship's hotel services and cargo heating and air conditioning. A 7500 lb/hr oil-fired package boiler provides emergency 150 psi pressure steam for heat and fire protection. Both the diesel generators and the package boiler are normally idle.

REACTOR SYSTEM MAINTENANCE AND CORE REFUELING

Hatches in the containment vessel permit replacement of the primary pumps, valves, instruments and wiring if such replacement should be necessary. Replacement of the reactor vessel, pressurizer and steam generators is not provided for. The boiler heads are designed, however, to permit access to the tube sheets for repair or plugging of leaking tubes. A program of preventive maintenance with respect to instruments, wiring and valves, with inspections every four months or more, should forestall failures in these components while at sea.

Refueling of the core can only be accomplished when the ship is perfectly steady, but may be carried out at any dock equipped with the necessary crane and means for carrying away the spent core. A large "transfer sleeve," about 12 ft. in diameter, is first lowered around the reactor head to rest on the lower flange. The control rods are then disconnected from the drives, the reactor head is removed, and the transfer sleeve is filled with water. The core hold-down assembly and upper grid are then removed and individual fuel elements withdrawn in an appropriate shielded cask. Finally, the control rods are replaced and a new core installed.

WASTE DISPOSAL PHILOSOPHY

The plant is designed to require no waste disposal during normal operation except venting of the gas in the waste tanks, and only disposal of the containment air if access to the containment is required at sea. Both the gas from the waste tanks and the containment air will be passed off through one of the masts which will be equipped with high-performance filters.

Disposal of the demineralizer resin in the primary purification system will be accomplished by replacing the demineralizers (in their shields) periodically. Disposal of the activated resins, which may contain fission products, may then be carried out at facilities designed for this purpose. Drainage inside the containment vessel will be collected in a drain tank.

The contents of either or both primary loops (exclusive of the reactor) may be drained to tankage installed for this purpose. This will permit repair work on a boiler or loop without having to dispose of its primary water.

Activated gases from the top of the buffer seal surge tank and the pressurizer will be withdrawn either periodically or on a continuous basis, whichever is more convenient.

In general, all waste disposal philosophy has been postulated on the following:

(a) The presence of fission products in the primary loop must not impair the operation of the plant in any way.
(b) Fission products will not be dumped to the sea.

PLANT SAFETY PROVISIONS

The safety provisions incorporated in this plant design are of six general types.

Reactor Safety

The reactor is designed to scram automatically from any one of seven causes: a short period below 10 percent power, excessive power, excessive drop in primary pressure, excessive outlet temperature, loss of power to primary pumps, loss of power to safety circuits, or loss of power to control rod drives.

Plant Functional Radiation Monitoring System

Radiation monitors are provided in various parts of the reactor system in order to detect functional defects and send appropriate alarm signals to the control center. The monitored items include the following:

(a) Fission product activity in the primary water.
(b) Activity in the boiler water as a result of a boiler leak.
(c) Activity in the intermediate water as a result of a leak in the letdown cooler or primary pump coils.
(d) Activity in the demineralizer exit water (to determine resin bed saturation).
(e) Activity in the containment air (to indicate possible leaks and also when purging has been sufficient for access).

Biological Radiation Monitoring System

Detectors will be located at various points around the secondary shield to monitor radiation dose rates so that the permissible levels of 0.5 R/yr in the passenger spaces and 5 R/yr in the holds and machinery compartments will not be exceeded. General air activity will also be monitored on deck, in the en-

gine room, in the control room and probably on the bridge.

Emergency Power and Backup Provisions

For normal operation, the power plant has several backup provisions. It has two independent boilers, each capable of furnishing almost ¾ of normal power; four independent primary pumps, each with two speeds; two 1500 KW turbine generators, each capable of sufficient power to maintain the ship's electrical load at reduced power; three demineralizers; two letdown coolers; two charge pumps; two intermediate heat exchangers; and considerable duplication in the instruments and controls.

Upon loss of reactor power, emergency power will be provided by the two 750 KW diesel generators which are started automatically, a 300 KW diesel generator top-side and a constantly-charged battery bank for vital nuclear instrumentation. The 7500 lb/hr package boiler will supply emergency steam.

Isolation

Another safety feature of this plant is that defective units, such as boilers, letdown coolers, pumps, demineralizers, control rod buffer seals, etc., that might contaminate the plant can be isolated.

Containment

The containment vessel is undoubtedly one of the principal safety features of the ship. The underlying containment philosophy is:

(a) Radioactive matter of sufficient intensity to impart 300 mr/wk shall be doubly contained such that simple rupture of the primary container will not cause rupture of the secondary container.
(b) The containment vessel shall be located where damage to it from collision of the ship will be least probable or severe.

In the occurence of brittle fracture in service, many failures have occurred under conditions of dynamic loading. These conditions have been duplicated in the laboratory, and the mechanism of failure under such loading has been considerably clarified by laboratory tests. However, instances of brittle fracture at static loads well under that necessary for plastic deformation have also occurred in service. It is not believed that general yielding in the root of a notch or discontinuity could have preceded initiation and propagation of the crack. Recent work by Mylonas, Drucker, and Brunton has been successful in producing, for the first time in the laboratory, conditions under which brittle fracture occurred at average stress levels as low as 36% of the virgin yield stress. This has been done by supporting two opposite edges notched steel plate, 10 inches square, with plastic hinges, then applying the tensile load in a direction perpendicular to the hinge line. The hinges serve to eliminate eccentricity of loading of the plate specimen in any direction out of the plane of the plate. Series of tests were run at low temperatures with the plates prestrained in both tension and compression. From the results, it was concluded that residual stresses at the notch roots were not of great importance in changing the net average stress of brittle fracture under such loading. However, the strain history of the metal at the notches appeared to be of great significance.

—from WELDING JOURNAL
October, 1958

The S. S. SANTA PAULA, delivered October 13, together with her sister ship, the S. S. SANTA ROSA, delivered June 16, are the newest additions to the passenger fleet and the first all new passenger ships in the American merchant marine since 1952. Owned and operated by the Grace Lines, the ships will operate between United States and Caribbean ports on a weekly schedule. The ships have a length of about 584 feet, displace nearly 19,500 tons and have a normal speed of 20 knots. The liners were built by the Newport News Shipbuilding and Drydock Company, and incorporate the most modern and luxurious features for passenger convenience. Spacious accommodations with completely air conditioned rooms and passenger spaces, and anti-roll stabilizers are indicative of the emphasis placed on passenger comforts. The single class accommodations provide for 300 passengers. All rooms are outside and each room is equipped with a private bath. The ship is outfitted with the most modern cargo handling gear, including a cargo elevator and provisions for handling palletized cargo. The 20,000 SHP is provided by two 10,000 SHP turbine propulsion units, operating at 600 psi and 850°F steam conditions.

* * * *

A preliminary design investigation into a hitherto untried method of reactor control has been conducted for the past 21 months at the Babcock and Wilcox Atomic Energy Division. The method, adapted to control of water moderated and cooled reactors, is termed the "Spectral Shift Method."

Designed to replace the control rod system, the new technique makes use of varying the proportions of light and heavy water in the coolant-moderator. Although several major problems remain unsolved, the advantages to be gained from this type of control have led to a continuing development program designed to overcome the obstacles.

One of the chief advantages expected is a longer core lifetime. This is expected to be achieved by extending the reactivity life of a core to as much as 40,000 megawatt days per ton of fertile material, more than twice as much as that expected from a normal pressurized water reactor. Another advantage is that which accompanies non-uniform loading of the core—a technique made possible by the elimination of most of the control rods. Non-uniform fuel patterns can result in a more even power distribution through the core volume, permitting a higher average power output from a given core size.

An initial high concentration of about 80 per cent heavy water, replacing the more expensive control rods, is used to control the excess reactivity in the new core. As the core ages, the concentration of heavy water must be reduced to permit criticality. The changing concentration of heavy water shifts the energy spectrum of the moderated neutrons. This principle is responsible for the name "Spectral Shift Control".

—from BABCOCK AND WILCOX press release

September, 1958

ROBERT TAGGART

NOVEL MARINE PROPULSION DEVICES

THE AUTHOR

who recently was the Technical Administrator of Reed Research, Inc., has just incorporated as Robert Taggart, Inc., to engage in consulting and marine research and acoustics. He received his Bachelor of Science degree in Naval Architecture and Marine Engineering from Webb Institute of Naval Architecture in 1942. He served with the Consolidated Steel Corporation, U.S. Maritime Commission, U.S. Army Transportation Corps, and the Navy Department prior to his employment with Reed Research in 1954.

EDITOR'S NOTE: This is the third article in this series by Mr. Taggart. Because of its length it has been divided into two parts. The second part will be published in the February 1959 issue of the JOURNAL. All of the references will be included with the second part.

INTRODUCTION

THE DEVELOPMENT of marine propulsion systems has been in process for centuries. By gradual and successive steps, these developments have led to the almost universal use of the screw propeller for all forms of marine craft. The process has not been without its share of digressions into many ideas and experiments with other methods of propelling waterborne vehicles. As in most technical fields which are attractive to the inventive mind, these digressions range from the semi-practical to the ridiculous.

Yet it is quite possible that among the thousands of ideas which have been advanced there may be some of distinct value which have been overlooked with the passage of time. It is possible that some of the unusual methods of propulsion advanced in earlier years were considered infeasible in their time because materials or machinery had not been developed which would render them practical. It is also possible that, in striving for greater propulsive efficiency, the potential of some types of propulsion systems for particular applications has been overlooked.

The ideas which have been generated run the gamut from the simplest mechanical adaptation of the paddle or oar to extremely complex forms of gaseous jet propulsion. They cover the utilization of novel sources of energy from the wind and the sea to abstract forms of power not yet fully understood. Some methods employ sound techniques, based upon the practical fundamentals of mechanics and physics, whereas others use techniques which are obvious contradictions of known laws. But among the many methods described on the following pages

may be one which can be employed effectively in some future installation for the propulsion of marine vehicles.

OSCILLATING BLADES

The oscillating blade is probably the oldest form of marine propulsion device. The hand and foot of a swimmer, the paddle and oar of a small craft, the sculling blade, and the various forms of duck-foot and goose-foot propulsive devices fall under this category. Although it is an ancient form of propulsion, it has generally been proved to be highly inefficient. Yet, from the basic principles involved have developed the paddle-wheel and the screw propeller.

The various forms of devices which attempted to duplicate the pedal motions of aquatic birds prior to 1800 have been described previously. As might be suspected, similar attempts have been made since that time. About 1826, a Mr. Nairn contrived a device which was something of a hydrodynamic improvement over those of Lord Stanhope, but which operated on the same basic principle. Two, four, or more levers were suspended over the sides of the vessel extending down to the plane of the keel. They were pinned above the waterline and were driven fore and aft in a pendulum-like motion. At the lower ends of the levers were attached hinged blades, as shown in Figure 1. These blades folded

Figure 1. Nairn's Oscillating Blades.

into a parallel position on the forward stroke and opened up on the aftwards stroke. The opening angle was restricted to between 140 degrees and 160 degrees by chains.

Later in the nineteenth century, a similar device was patented called the *side fan propeller.* As shown in Figure 2, this consisted of a series of umbrella-like devices fitted on a reciprocating rod. Rods through the forward and after umbrellas were fitted to a linkage so that as the after umbrellas moved in one direction the forward umbrellas moved in the opposite direction. The umbrellas opened on the aftwards stroke and closed on the forward stroke. Due

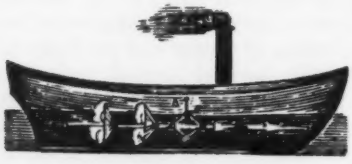


Figure 2. Side Fan Propeller.

to the phase difference in the forward and after umbrellas, a continuous propelling action was achieved.

In 1881, a 30-foot steam yacht was built with a 36-inch blade hung under the overhang of the stern. The blade was again designed so that it closed up on the forward stroke and opened on the aftward stroke, *virtually pushing the water astern.* The blade oscillated at the rate of 120 cycles-per-minute. Although it was *calculated that extraordinary speed* would be obtained, there seems to be no record of its actual performance.

Some inventors attempted to follow the example of John Fitch in actually duplicating the motion of oars. One such invention was the ship *Propeller,* built in England in 1840. By an arrangement of cranks and levers, the *Propeller* moved a series of vertical blades on each side in an aftwards direction and then, retracting them for the return stroke, would again immerse them in the water. It was said that, from the side, this vessel resembled a grasshopper.

A very interesting mechanism was the oscillating blade, patented by William Lanphier Anderson of Norwood, England, in 1853. This device is shown in Figure 3. As the crank, H, was rotated by the gear train, the blade arm would pivot about a spherical bearing in the side of the ship and the blade would travel in a circular arc. The mechanism was so designed that the blade would be perpendicular to the stream on the aftwards stroke and would feather on the forward stroke.

In the next year, 1854, Charles de Berque, of London, took out a patent on another type of oscillating fin, shown in Figure 4. This fin was carried in a type of paddle-box on the side of the ship. By a combination of a crank and a linkage, the fin oscil-

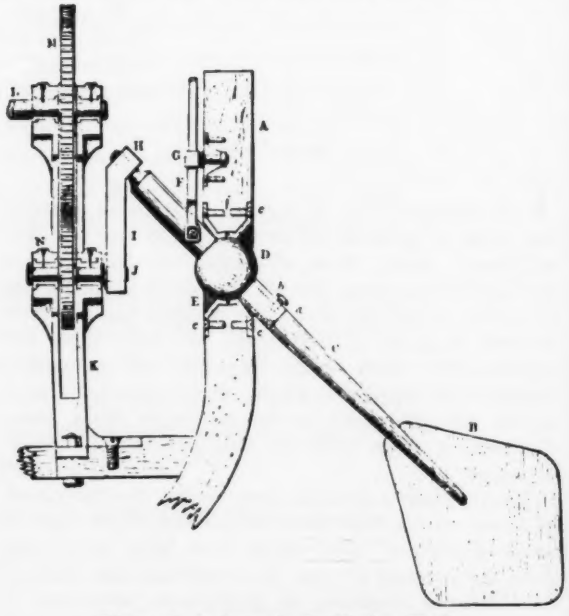


Figure 3. Anderson's Oscillating Blade.

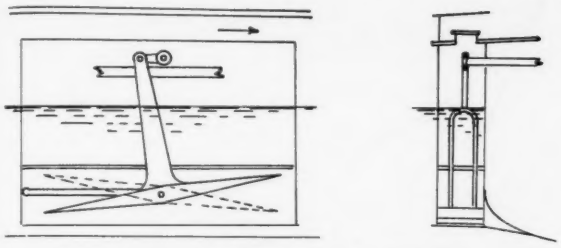


Figure 4. The Oscillating Fin of de Berque.

lated in a manner which pulled the water in at the forward end, trapped it between the fin and the bottom of the box, and then forced it aft. The inventor claimed this to be a great improvement over the paddle wheel.

Also in 1854 Auguste Edward Loradoux Billford of London patented a mechanism for sculling a ship forward by means of port and starboard paddles. The motion of the paddles resembled that of the ancient galleys, that is, the principal oscillation of the paddle shaft was outboard and inboard with the shaft being rotated so as to give forward thrust throughout most of the travel of the paddle.

A similar device was developed by J. E. Allen of the David Taylor Model Basin about fifteen years ago and a small model was built to test the principle. This model utilized more advanced forms of hydrofoils to provide the propelling force and demonstrated that reasonable propulsive efficiency could be achieved in this manner.

Generally speaking, the oscillating fin represents a proven method of ship propulsion. It does, however, have the disadvantage of requiring somewhat intricate mechanism to attain the desired motion, which is bound to reduce the mechanical efficiency and the reliability of the device.

FISH PROPULSION

Probably since the dawn of history man has hoped to emulate the propulsive methods utilized by birds in the air and fish in the water. Yet the simple simulation of these muscular contortions by mechanical means has always been a decided stumbling block. Over the years, however, many inventors have tried to accomplish these motions in the hopes of arriving at a more efficient propulsion system.

Fish and sea mammals provide examples of many different forms of propulsion. Some fish use a flapping side fin for propulsion at low speeds. The squid ejects liquid from its tentacles, in a form of jet propulsion. The manta ray undulates its wings in a manner simulating the soaring action of some types of birds. And the aquatic birds simply paddle with their feet.

The majority of the fishes and sea mammals have a two-speed propulsion system. At low speeds, their tails oscillate back and forth or up and down—an action similar to sculling. But at higher speeds their entire muscular system comes into play in an undulating action which starts well forward on the body and increases in amplitude toward the tail. The drawings of Figure 5, taken from successive photographs of a swimming eel, demonstrate this action in exaggerated form.

Whether simulation of fish propulsive methods can ever produce an efficient form of propulsion for man-made craft is still somewhat questionable. There are two opposing basic philosophies on this question which are well-illustrated in the following

Figure 5. Successive Motions of a Swimming Eel.

quotations from Dr. Manfred Curry and from Dr. Ir. J. D. Van Manen, of the Netherlands Ship Model Basin.

"In Nature we find the alternating flow principle used, in preference to the continuous flow principle, by living creatures for their propulsion in air and water. Nature, in general, goes far beyond all human conception in the fitness of her designs, and for this reason we may well assume that the method of propulsion used by birds and fishes is not accidental but was evolved on account of the very high efficiency of the vibratory principle.

"Vibration not alone saves energy but it enables the individual animals to couple themselves together through resonance and to utilize the powers thus liberated in their field of excitation. Doubtless Nature could have utilized a rotary movement for the propulsion of her creatures if she had so desired. In fact, the outer edge of a bird's wing does move in circle-like paths. If Nature preferred the alternating flow principle for animals living either in air or water, this principle of propulsion must possess many advantages with reference to efficiency as well as to transmission of energy." . . . Curry

"Anyone who is confronted with the problem of propulsion occasionally asks himself the question: Why should not we use the system practised by fishes, i.e. that of executing periodically repeated strokes? The answer to this question is in all probability implied by the fact that the technique of the employment of rotating parts, such as shafts, wheels, etc., has, from the viewpoint of efficiency of motion, obtained a lead over nature. A fish, just like a man on a bicycle, might advance in a more efficient manner with the aid of a screw propeller which would derive its rotation from strokes carried out with the tail. When taking a retrospective view of nature for solving our technical problems we should realize, however, that living beings form a totality to which considerations of efficiency can scarcely be applied." . . . . Van Manen

Despite arguments against its practicability, fish propulsion has been a subject of interest to inventors for many years. In 1848 Robert Fowles of North Shields, England patented the device shown in Figure 6. This consisted of one or more horizontal blades installed at the stern of the ship. The blade arms were pinned to a fixed vertical rod and to a reciprocating vertical rod. The reciprocating motion drove the blades in an oscillating fashion, thus propelling the ship. The blades were constructed of steel ribs covered with gutta perche or vulcanized india rubber to give them flexibility which would simulate a fish tail. Fowles also suggested placing the blades in a vertical plane so that they could be used in steering the ship.

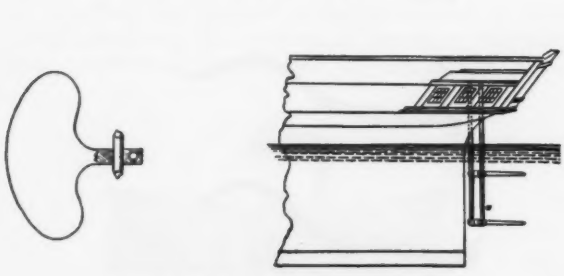


Figure 6. Fowles Fish Tail Propeller.

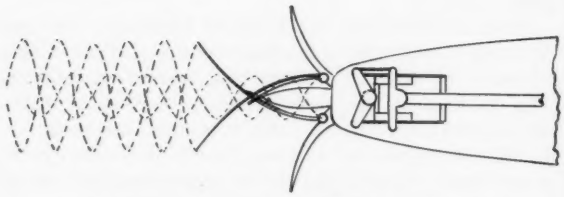


Figure 7. Johnson's Vibrating Propeller.

A very intriguing method of fish propulsion for boats was patented by William H. Johnson of Springfield, Massachusetts in 1862. Figure 7 shows the general form of the installation and details the oscillating mechanism and the track of the undulating fins. A handcrank working through a gear train oscillated the connecting rod in a fore and aft direction. Through a linkage mechanism, the crosshead on the connecting rod rotated two concentric shafts in opposite directions. To these counter-rotating, oscillating shafts were connected the propulsion devices. These devices were made up of rigid bifurcated arms to which were pinned flexible blades. As the arms swung in their oscillating arcs, the flexible blades would swing from one leg of the fork to the other. It was claimed that the resulting sinuous motion of the flexible blades would enable them to present a large proportion of effective propelling surface to the water.

Dr. Arno Boerner of Dresden, an inventor of internal combustion engines, turned his attention to the relationship between fish and boats in 1927. He came to two erroneous conclusions regarding fish propulsion. First, he concluded that a fish swims by pumping water from its gills. With this idea in mind, Boerner built the Gill Boat shown in Figure 8. In the bow, he installed a large water turbine which sucked water in at the stem and ejected it through forward openings, which were intended to simulate gills. Secondly, Boerner concluded that the reason a fish could swim so rapidly was because of a rough skin which caused a series of vortices with forward acting components. He roughened the bottom of his

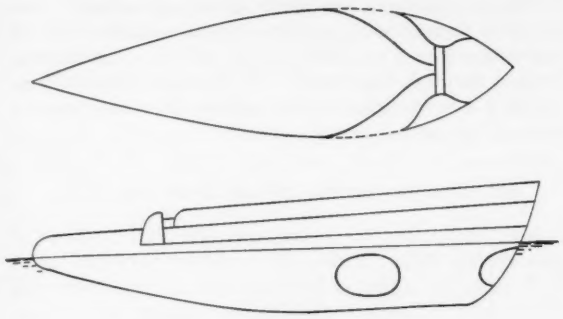


Figure 8. Boerner's Gill Boat.

boat to attempt to gain a propulsive force from these vortices. His boat, named the *Forelle*, was 30-feet long with a beam of 4-feet powered by a 6 horsepower engine. Trials were run in the Elbe River and, with the characteristic vagueness of inventors evaluating the products of their trade, Dr. Boerner claimed to have made a speed three times that which would be expected for 6 horsepower. Yet, also characteristically, no more was ever heard of this invention.

In 1929, the application of a fish propulsion technique to the propulsion of boats was discussed by Dr. Manfred Curry, the famous author and authority on sailing craft. In a forthright and logical dissertation, Dr. Curry asserted that if nature adopted the vibratory principle in the flight of birds and in the swimming of fishes, it was not accidental but based on the high efficiency of the principle. The construction of the boat shown in Figure 9 was recommended. This boat had a propulsion system which consisted of an oscillating cross head driven by foot power through a linkage. Connected perpendicularly to the cross head was the forward end of a flexible fin. Oscillation of the cross head caused an angular oscillation of the forward end of the fin. Due to the fin's flexibility and the transverse resistance of the water, the fin would be forced into an undulation simulating the motion of the afterbody and tail of a fish. Dr. Curry mentioned the construction of a similar boat by a German, Hans Schramm, which was motivated by an engine.

This particular form of fish propulsion is generally termed *flexible fin propulsion*. It has proved to be quite practical for certain applications. During World War II, craft were fitted with fins of this type for operation by Commandoes on the rivers of Burma. The design was similar to that proposed by Curry, but with provision for multiple pedallers.

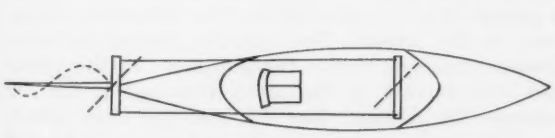


Figure 9. Curry's Flexible Fin.

These craft had two distinct advantages. First, they utilized the leg muscles of the oarsmen, rather than the weaker arm muscles. Second, there was practically no noise generated by the undulating motion of the fin.

Lack of noise is a characteristic which may well bring about the use of fish propulsion mechanisms in the future. For silent operation, it has no peer. Measurements made by the Woods Hole Oceanographic Institute, in the tanks at Marineland, Florida, of the noise of porpoises produced some rather startling results. Aside from vocal noises and splash noise made when they broke the surface, these mammals were amazingly silent. Even in the fastest starts and maneuvers, there was no indication of cavitation or cavitation noise. If these characteristics could be attained in modern submarines and underwater weapons, a great advantage in undersea warfare would be gained.

DRAWING PROPULSIVE POWER FROM WAVES

The tremendous amount of energy existing in the wave motion of water particles at sea has always been intriguing to the inventor of marine propulsion devices. Conversion of only a small amount of this energy to forward thrust would represent a great advance in marine science.

Carl G. E. Hennig obtained a patent in 1874 for a craft which employed inclined planes at the bow and stern for wave propulsion. A more interesting and successful application of this same principle was developed by Herman F. L. Linden, in Naples, Italy, in 1898. Linden used the wave motion for a source of power and flexible fins for converting this power to propulsive thrust.

His first boat built on this principle was 13-feet in length and was named *Autonaut*. Horizontal fins were mounted on the bow and stern as shown in Figure 10. The fins were 20-inches long by 10-inches wide and tapered from 0.07-inches in thickness at the forward end to 0.01-inches at the after end.

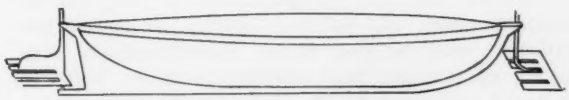


Figure 10. Linden's "Autonaut."

The pitching motion of this boat was sufficient to undulate the fins and thus give a propulsive forward thrust. The *Autonaut* made a speed of three knots free running, against both wind and sea, and more than two knots with a plank in tow. Linden built several of these boats of greater length than the *Autonaut*. The largest was 24-feet in length and made a speed of four knots.

A New Hampshire inventor, Sotirios Kalfas of Manchester, patented another variation of the wave propelled boat, in 1936. Figure 11 demonstrates the principle involved. As the boat pitched in a seaway,

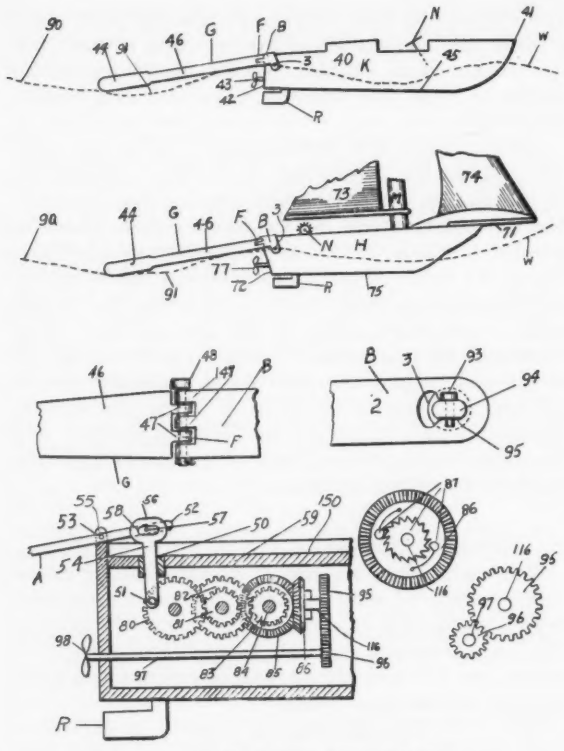


Figure 11. Kalfas' Wave Propelled Boat.

the hinged paddle moved up and down relative to the boat. Through a gear-train, ratchet, and pawl this relative oscillation was converted into a continuous rotary motion which drove a screw propeller.

A much more ambitious plan was described by James Graham of New York City. In 1902 he obtained a patent on an automatically propelled multiple hull vessel. This vessel was articulated both longitudinally and transversely as can be seen in Figure 12. The hull consisted of four longitudinal sections which were hinged together to permit undulation with bow-on waves. Transversely it consisted of a main hull section mounted between buoyant pontoons which were hinged to the main hull to provide a relative movement under the action of transverse waves.

The hinges between the longitudinal sections consisted of pin joints at mid-depth. At deck and bottom levels were rows of pistons secured by ball joints to one section, moving in cylinders affixed to the adjoining sections of the hull. These cylinders contained hydraulic oil which was pumped by the undulatory pitching action of the hull.

Between the outer pontoons and the main hull were roller joints about which the floating sections pivoted. To these joints were attached chains which actuated hydraulic cylinders.

The *tremendous* hydraulic power generated was pumped to hydraulic motors which, in turn, rotated the seven screw propellers with which the ship was moved through the water. Mr. Graham wisely provided a conventional steam power plant for operation of his vessel in calm water.

DRAWING PROPULSIVE POWER FROM THE WIND

For centuries the wind was the sole source of power for ships. From the crudest form of craft with skins of animals hoisted aloft, to the towering masts and billowing sails of the clipper ships, man had continually striven to gain better advantage of this force which was readily available to drive his craft through the waters. Yet, even at peak efficiency, it was somewhat questionable as to whether the greatest advantage was being obtained.

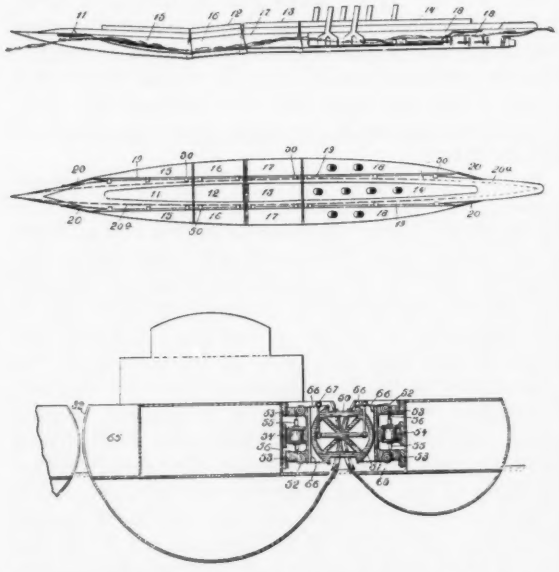
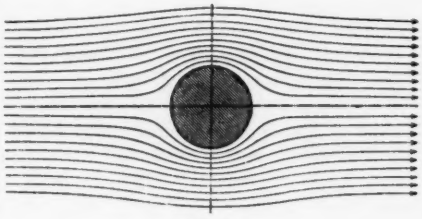


Figure 12. Graham's Automatically Propelled Vessel.

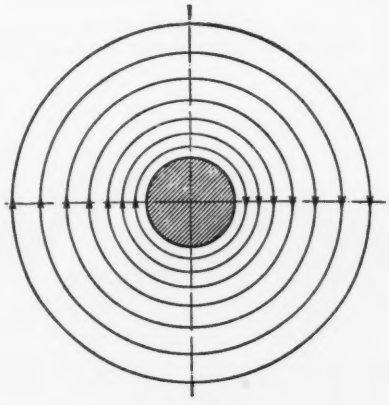
The basis for a major step forward in the use of the wind for propulsion was a physical phenomenon uncovered by Gustav Magnus in 1852 when he was a Professor of Physics at the University of Berlin. Magnus was also serving as an instructor in the Artillery Academy and became interested in the problem of cannon balls being deflected in a lateral direction from their true ballistic path in flight. He performed a series of experiments, using a brass cylinder rotating in a wind stream and measuring a lateral force on the cylinder in a direction orthogonal to the stream. The cylinder tended to move towards the side where the periphery of the cylinder was moving in the same direction as the air current. He realized that this was the force which had been deflecting projectiles and, although he did

not measure its magnitude, he is rightly credited with the discovery of the Magnus Effect.

This effect can be understood by reference to the various sketches in Figure 13. Figure 13A shows the streamlines of flow around a fixed cylinder. In Figure 13B is shown the imaginary circulatory flow of a current flowing around the cylinder in concentric circular paths. Combining these two flows results in the theoretical pattern of Figure 13C. Considering the theoretical flow of air past a rotating cylinder, the flow pattern of Figure 13D is established which results in a high velocity flow on the upper side of the cylinder and a low velocity flow on the lower side of the cylinder.



13-A



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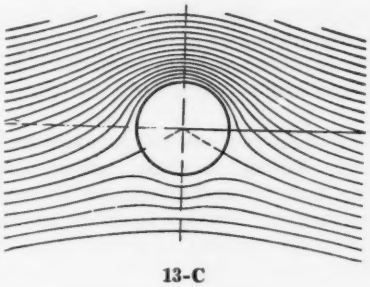

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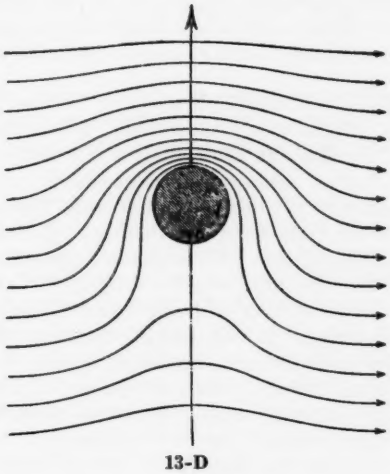

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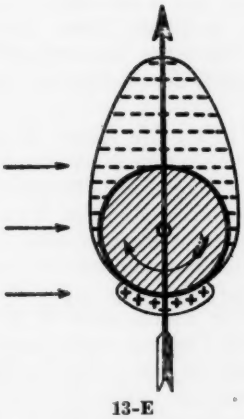

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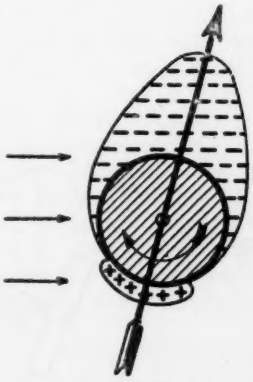

13-F

Figure 13. Description of the Magnus Effect.

This variation in velocity on either side of the cylinder results in a variation in pressure which is inverse to the velocity variation in accordance with Bernoulli's theorem. The resulting pressure distribution is shown in Figure 13E, where the plus signs indicate the relative pressure increase and the minus signs indicate the relative pressure decrease. The arrow indicates the force which is generated due to this pressure difference. In the actual case, when frictional effects are taken into account, the conditions differ slightly, as can be seen in Figure 13F.

In 1877, Lord Rayleigh wrote a treatise: "On the irregular flight of a tennis ball," in which he described the Magnus Effect; but it was not until 1910 and 1911 that actual measurements were made of the magnitude of the forces involved. These experiments were conducted by Professor Lafay, in the Physical Laboratory of the École Polytechnique.

Professors Prandtl and Foettinger, at Goettingen, became interested in this effect in 1918 and the latter encouraged a somewhat dubious Professor Guembel to construct a model propeller illustrating the Magnus Effect, which is shown in Figure 14. The application is fairly clear: The rotation of the drive shaft rotates the two cylindrical *propeller blades* in opposite directions. When inserted in a wind stream, a torque is developed by the opposing forces and the propeller turns. Both Professors Foettinger and Guembel considered this as an interesting laboratory demonstration but of no practical value.

In the early 1920's the sailing ship was swiftly passing out of the picture as far as commercial maritime use was concerned. To many, this ending of an era was a source of regret. To some, like Anton Flettner, the thought of losing the wind as a source of propulsive power was untenable. His aim was to utilize the wind in a more efficient manner than ever before in the centuries of ships under sail.

In 1922, Anton Flettner took over the management of the Institute voor Aero-en-Hydro-Dynamick in Amsterdam, Holland. In collaboration with the Aerodynamic Experiment Station in Goettingen

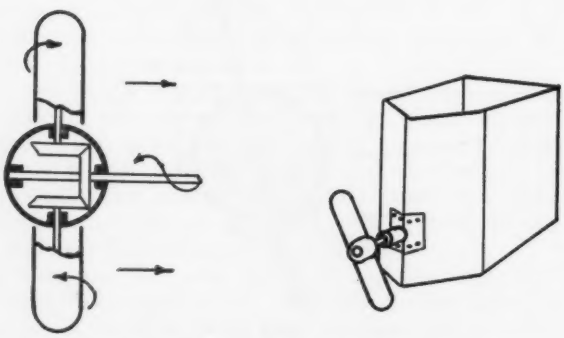

Figure 14. Guembel's Model Magnus Propeller.

numerous experiments were conducted on methods of obtaining power from the wind. After several experiments with metal airfoils as sails, Flettner conceived the idea of utilizing the Magnus Effect. He pictured a ship with large, vertical cylindrical rotors rising from the deck which would develop a forward thrust from the velocity of a transverse windstream. In 1923 he patented the idea.

Flettner went with his idea to Prandtl, Betz, and Ackeret, at Goettingen. After a series of model experiments had been performed, the three-masted topsail schooner *Buckau* was obtained for an experimental installation. It had been decided to use two rotors with a projected area one-tenth that of the sail area of the *Buckau,* which was 8,500 square feet. The cylinders were made of 3/64-inch steel sheets and were internally stiffened and braced. The rotors were 43-feet high and about 10-feet in diameter. They were driven by two reversible 220-volt, direct current, shunt motors of 11 kw operating at 750 RPM.

Conversion of the *Buckau* was completed in 1924. A sketch of her before and after conversion is shown in Figure 15. Both dock and sea trials were

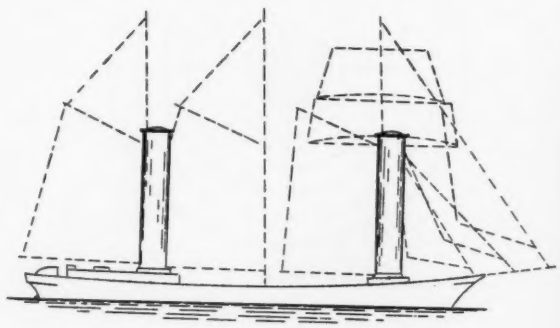

Figure 15. The *Buckau* Before and After Conversion.

conducted with extremely satisfactory results. She made several exhibition trips around Europe and the British Isles. Re-christened the *Baden-Baden* she made the Atlantic crossing to New York in 1926. Subsequently, a naval vessel, the *Barbara,* and several small yachts were built on the rotor ship principle.

Anton Flettner was not only a scientist and inventor but he had the facilities, patience, and talent to follow his inventions through to a successful commercial conclusion. Yet he too had his difficulties. He said:

"Only he who, by nerve-wracking struggle extending over years, has almost forced upon the engineering community an idea of his own which is considered impracticable, can understand how an inventor feels when, after his efforts have led to a certain degree of success, he reads criticism of his work in which a competent engineer proves that the invention represents nothing but a very obvious technical

operation. An incredible amount of perseverance, nervous energy, and faith, is required to carry the fight for a new cause through to the end. Woe to the inventor who cannot find the iron will for this hard struggle. Only too often the history of inventions tells us of most ingenious originators of truly great ideas who have succumbed to this struggle."

While Flettner was engaged in the construction of his rotor ship, others were also interested in the utilization of wind for propulsive power. Based upon the earlier proposal of M. Duguet, an engineer named Constantin designed and built a windmill-propelled boat called the *Bois-Rose*, shown in Figure 16. In September 1924, Constantin ran trials on this boat in the Seine. It had a 30-foot diameter, two-bladed, Levasseur propeller. The propeller was mounted on a lattice framework, which was free to pivot in all directions, on the deck of the boat and, through shafts and gearing, drove a screw propeller. This boat was said to have made a speed of 5 knots with a wind velocity of 14 knots.

Figure 16. Constantin's "Bois-Rose."

At the time Anton Flettner went to Goettingen, Prandtl, Betz, and Ackeret had been working on the phenomenon of boundary layer control. This field involved creating some sort of disturbance in the thin layer of air or water created along the surface of a body by flow past it. The rotation of Flettner's rotor was actually one form of boundary layer control; there exist many other methods.

One boundary layer control method which is fairly well-known is the *slotted wing*, shown in Figure 17. The slot, *2*, extending along the span conveys the relative flow of air below the wing, *1*, to its upper side. This modification permits a much greater lift at low speeds than would normally be possible. However, the increased lift is accomplished by a correspondingly greater drag.

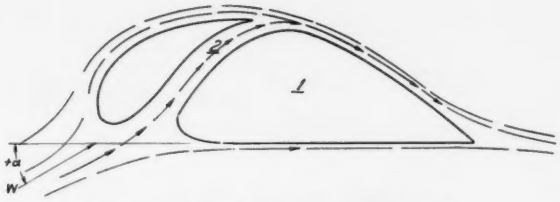

Figure 17. Conventional Slotted Wing.

The Goettingen scientists attempted to use a slotted wing in the place of a sail on a small boat, but were notably unsuccessful in their experiments. They then tried several other boundary layer control and energization methods. These included sucking air from the boundary layer on one side of an airfoil and also of pumping air into the boundary layer. Using an advanced boundary layer energization technique, Theodore Von Karman has recently constructed a small sail boat with a porous mast and movable flap for a sail.

Although these techniques may never prove useful in providing wind propulsion as a substitute source of power for ships, they do offer attractive possibilities in other aspects of ship propulsion and control. Screw-propellers utilizing the basic principle might be developed with very high thrust for particular applications. Control surfaces, such as rudders, diving planes, and anti-rolling fins might well use these techniques to obtain better performance.

NOVEL SOURCES OF PROPULSIVE POWER

The swift flowing waters of the River Elbe in Germany were overcome in the 1880's by a novel propulsion device utilized by steam tugs. A heavy iron chain was laid on the bottom of the river with its ends secured at the terminals of the navigable part of the channel. The tugs were fitted with winches through which the chain would pass. Thus the tugs could tow a line of barges by winching their way along the chain. Sufficient slack was allowed in the chain to permit the tugs to use their rudders to navigate around other ships. The *chain steamers* were double ended, with rudders fitted bow and stern, to permit them to operate equally well up and down stream. About 25 miles of chain were laid in the the Austrian section of the Elbe and about 390 miles were laid in the German section of the river.

The third in the line of American capitalists bearing the name of John Jacob Astor was also an inventor. When not occupied with the family business interests, he turned his talents to the invention of

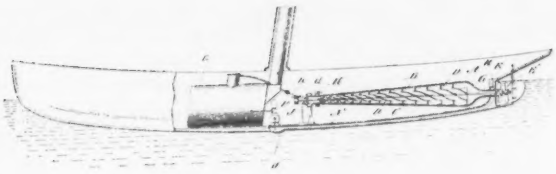

Figure 18. Astor's Turbine.

such devices as a bicycle brake, a pneumatic road improver, and the steam turbine patented in 1902, which is shown in Figure 18.

The turbine was a single-stage screw-type in which both the inner rotor and outer casing were free to revolve. The revolving rotor and casing drove contra-rotating propellers. Steam was admitted to the forward end of the turbine and exhausted at the after end. Another novel feature in this system was the keel condenser through which the steam passed in its return forward to the boiler.

In spite of John Jacob Astor's ample financial means it is not believed that his turbine was ever constructed, probably because of the obvious mechanical difficulties involved in such a machine.

In 1927, Louis Sher of Philadelphia, Pennsylvania, applied for a patent on a device which was reminiscent of the *Great Principle* of M. Genevois, in 1759. Sher's idea was to utilize a spring energy storage system to operate an air screw propeller, as shown in Figure 19. The energy was stored in the springs by means of a prime mover which operated on a series of levers and ratchets, thus distending the group of parallel springs. Provision was also made in this patent for using a catamaran type hull and a variety of other propulsive units driven by springs.

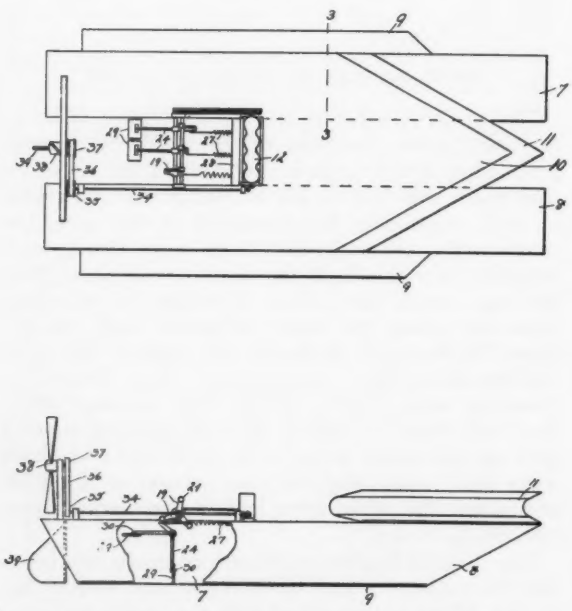

Figure 19. Sher's Spring Power Propulsion Device.

No discussion of the history of marine propulsion would be complete without including the ideas of the most fertile imagination of the nineteenth century. The designs of this man were not set down in the form of drawings but in the form of words—words so exciting that they have stirred the hearts of both the adventurous and the scientific. For this man was both an adventurer and a scientist, an inventor without patents, a deep student of nature and of the physical laws which govern the creations of man. He was Jules Verne.

The source of power for Verne's submarine *Nautilus* can best be described in the words of Captain Nemo, from "Twenty Thousand Leagues Under the Sea":

> "There is a powerful agent, obedient, rapid, easy, which conforms to every use, and reigns supreme on board my vessel. Everything is done by means of it. It lights it, warms it, and is the soul of my mechanical apparatus. This agent is electricity ... my electricity is not everybody's. You know what sea-water is composed of. In a thousand grammes are found 96½ per cent water, about 2⅔ per cent of chloride of sodium; then, in a smaller quantity, chorides of magnesium, sulphate of magnesia, sulphate and carbonate of lime. You see, then, that chloride of sodium forms a large part of it. So it is this sodium that I extract from sea-water, and of which I compose my ingredients. I owe all to the ocean; it produces electricity, and electricity gives heat, light, motion, and, in a word, life to the Nautilus."

And, on her machinery and propulsion system:

> "I use Bunsen's contrivance, not Ruhmkorff's. Those would not have been powerful enough. Bunsen's are fewer in number, but strong and large, which experience proves to be the best. The electricity produced passes forward, where it works, by electro-magnets of great size, on a system of levers and cog-wheels that transmit the movement to the axle of the screw. This one, the diameter of which is nineteen feet and the thread twenty-three feet, performs about a hundred and twenty revolutions in a second."

This was written in 1869, before the *invention* of the electric motor, the primary battery, the incandescent lamp, and electric heaters. Only now, with the *USS Nautilus* (SSN 571) and its successors are we beginning to approach the fulfillment of the imaginative writings of this nineteenth century genius.

ACKNOWLEDGEMENT

The literature search on which this article is based was initiated under Contract DA-44-177-TC-409 with the U. S. Army Transportation Corps.

EDITOR'S NOTE: All of the Author's references for this article will be printed at the end of Part II which will be published in the February 1959 issue of the JOURNAL.

EDWARD V. LEWIS

POSSIBILITIES FOR REDUCING SHIP MOTION AT SEA

ACKNOWLEDGEMENT

This is a slightly abridged version of a paper presented at the Second Navy Science Symposium in San Diego, California, in March 1958. The paper was prepared under contracts with the Bureau of Ships and the Office of Naval Research. The author is Edward V. Lewis, of the Experimental Towing Tank, Stevens Institute of Technology.

INTRODUCTION

For many reasons the reduction of ship motions at sea is a desirable goal for both merchant and naval vessels. Usually the velocities and accelerations associated with motions in waves are particularly objectionable, and, for a given frequency of encounter, the maximum values of the velocity and acceleration components are directly proportional to the amplitudes of roll, pitch, heave, etc. Thus, motions may have a serious effect on the functioning of machinery, instruments, and gun-control devices. Furthermore, accelerations are believed to be primarily responsible for seasickness and frequently cause shipboard accidents to personnel. The reduction of motions, consequently, can be expected to improve the functioning of any ship.

The subject of ship motions at sea cannot be considered separately, however, from course and speed as motions invariably can be reduced by changing course or reducing speed. Since most ships at sea are usually trying to get somewhere as quickly as possible, either of these steps is undesirable. Hence, the greatest value of any measures for reducing ship motion may lie in attaining higher speed or more direct course in storm seas. Within the same upper limit of accelerations for safe and efficient ship operation, a vessel with better motion characteristics can either maintain higher speed or a more desirable course than another vessel in the same storm area.

Other more indirect effects of ship motion on speed and performance in waves also should be considered. For example, in the case of a fully loaded cargo ship, shipping of water in head seas is a more serious threat to its safety and speed than motions or accelerations. Hence, an increase in freeboard or perhaps a modification in the forward flare may improve performance without making any change in the amplitudes of motion. Alternatively, a ship can be designed to cope with large quantities of water over the bow, as in the case of the new Canadian destroyer-escorts of the St. Laurent class. This involves a clean foredeck, an effective breakwater for deflecting water overside, additional strength in the underdeck structure, and—in the case of cargo ships—strong water-tight steel hatch covers forward. Such a ship could operate safely with more severe motions than a conventional ship.

In other cases, particularly when ships are not fully loaded, the critical factor in sea behavior may be slamming. It is significant that a comparatively fine ship with well-rounded bottom sections forward may slam less than another which actually pitches more violently. The former ship, consequently, may be able to maintain higher speeds than the latter in the same rough sea.

Nevertheless, in the cases discussed above, a reduction in amplitudes of motions would improve conditions by reducing both shipping of water and slamming, thus permitting higher speeds. The reduction of ship motions is, therefore, for many reasons an important goal for both commercial and naval vessels. Except perhaps for small hydrofoil craft which can rise clear of the surface waves and submarines which

can go below them, research has not yet revealed any panacea by which ships can be made to go at high speeds in rough seas without objectionable motions of some kind. What can and cannot be done in this direction is the subject of the present paper.

CONTROL OF ROLLING

Ship motions at sea are usually resolved into six components for study: the angular motions of roll, yaw, and pitch; the translatory motions of heave, surge, and sway. Some are more troublesome than others, but when one or two are reduced the others are apt to become more noticeable. Rolling has received particular attention ever since the days when steam replaced sails and the steadying effect of canvas was lost. Because of the fact that the forces involved in rolling are small, it has proved feasible to reduce these amplitudes drastically by the use of various anti-rolling devices. The simplest device is the bilge keel, which has been generally accepted in shipbuilding for many years. Its effectiveness can be explained on the basis of the theory that irregular storm seas contain many regular component waves of a wide range of periods superimposed on one another. Like a musician's tuning fork, a ship will respond much more violently to a particular wave period (or frequency) than to any other. This period is the natural rolling period of the ship. Consequently, most serious rolling occurs in the natural period of the ship and, therefore, may be classed as synchronous rolling. In any oscillating system, simple damping is always effective in reducing synchronous oscillation. This explains the reason for the effectiveness of the bilge keel: by increasing the damping of roll, it gives a marked reduction in serious synchronous rolling.

For moderate and high-speed vessels a much more effective method of reducing rolling appears to be controllable fins. Such devices are so effective that the rolling problem appears to have been solved in principle, except for low-speed vessels for which an activated tank or gyroscope system is applicable (see Figure 1). But this does not mean that further research is not needed, as roll control is not perfect. There are problems of the inter-action or coupling of other motions, particularly yaw, of obtaining increased effectiveness with reduced weight and cost, of avoiding structural failures of fin shafts, etc. Furthermore, the system for the control of roll needs to be co-ordinated with the steering system.

It is interesting to find that the more effective anti-rolling devices become, the more noticeable is the lateral motion of sway. When the beam of a ship is small in relation to the length of a wave, the ship partakes of the motion of the wave particles. Side sway is caused by the lateral component of the orbital wave motion shown in Figure 2. To cope with sway, Chadwick[1] called attention to the advantage in redefining

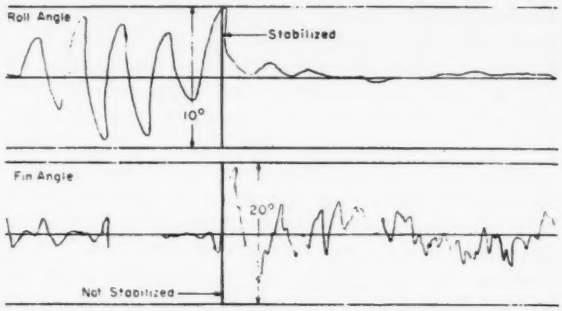

Figure 1. Effect of Stabilizing Fins on Roll, Wind Force 4, Following Sea, Ship Speed 20 Knots.[6]

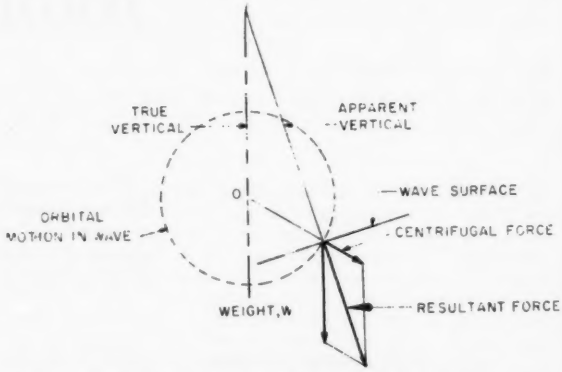

Figure 2. Forces Acting on a Small Body in a Wave.

the objective of roll stabilization in terms of stabilizing to the "apparent" rather than the real vertical ("Scheinlot" control in the German literature). As shown in Figure 2, the apparent vertical is the line of action of the resultant of the centrifugal and gravity forces, and in long waves this resultant is perpendicular to the wave slope. This simply means that if a ship is held to the apparent vertical it will have a sufficient heel angle at all times so that the thwartship component of gravity acting on any part of the ship or person aboard will exactly balance the lateral acceleration due to sway. An observer on board the ship then will *feel* upright because the lateral accelerations will not be noticeable. Only the vertical accelerations due to heaving and pitching (and small surging accelerations) will be apparent. Except for very short wave components, in which the true vertical is a better reference, apparent vertical control can be applied generally to roll stabilizers when comfort is the criterion. For some military applications the true vertical may be needed, and in this case the effect of sway cannot be avoided.

CONTROL OF OTHER MOTIONS

Considering next the use of rudders to control angular motion in the horizontal plane (yawing), the main objective of the helmsman or the automatic

pilot usually is to maintain an average course for the ship without attempting to limit greatly the periodic yawing in response to the encountered waves. There appears to be little doubt that present-day steering gears and automatic control devices could be improved in this respect. Furthermore, yawing undoubtedly could be reduced if this were considered desirable or necessary. This would require a quicker rudder response and a more refined control system similar to that used for anti-rolling fins, as described by Schiff and Gimprich[2]. Because of the strong coupling between roll and yaw, particularly in quartering seas, anti-rolling fins have a strong effect on steering. In fact, a ship actually can be steered by the use of anti-rolling fins alone. It would seem that the goal should be a single control system for the lateral motions of roll, yaw and sway, operating side fins and stern fin (rudder) in the most effective manner.

Of the modes of motion in a longitudinal plane, surge involves comparatively small accelerations and usually is not considered objectionable, except perhaps from the point of view of propulsive efficiency. If control were to be attempted it would involve tremendous margins of propulsive power, quick response of power plant to demands for rpm change, and a control system based on longitudinal motions and acceleration. Nothing of this sort is known to be under consideration at the present time.

Heaving and pitching motions are the most difficult to overcome and are the most serious problem, since they involve large vertical accelerations and may cause shipping of water, slamming, and propeller racing. Furthermore, when other motions are controlled or reduced, they become even more noticeable. Consequently, the balance of this paper will deal exclusively with these "symmetrical" modes of motion, under the assumption that methods are available for solving the other motion problems.

To what extent can special devices, either active or passive, be effective in reducing heave and pitch? It will be easier to answer this question after considering these motions in a more general way and discussing the effects of hull proportions and form.

NATURE OF PITCHING AND HEAVING

Ships are comparatively long, slender bodies, and the waves they encounter have lengths of the same order of magnitude—some shorter, some longer. Hence, the forces involved are large, and no one has even suggested that one try to reduce pitching or heaving to near zero, as has been attempted in the case of rolling. However, there is a possibility of reducing the motions significantly and improving the phase relationships between motions and waves.

Research in regular head seas has brought three important facts into prominence.[3] (1) the amplitudes of motion are greatest in the vicinity of synchronism between the period of encounter and a ship's natural period of oscillation (see Figure 3),

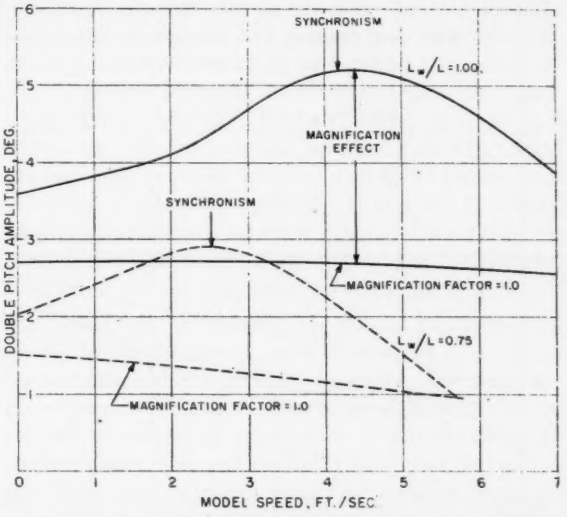

Figure 3. Calculated Pitch Amplitudes for Destroyer Model in Regular Waves of Height = 1/48 Model Length.

Amplitude peaks are shown at speeds near synchronism (i.e., at tuning factor = 1.0). The similarity between this figure and the familiar magnification factor diagram of vibration theory should be noted.

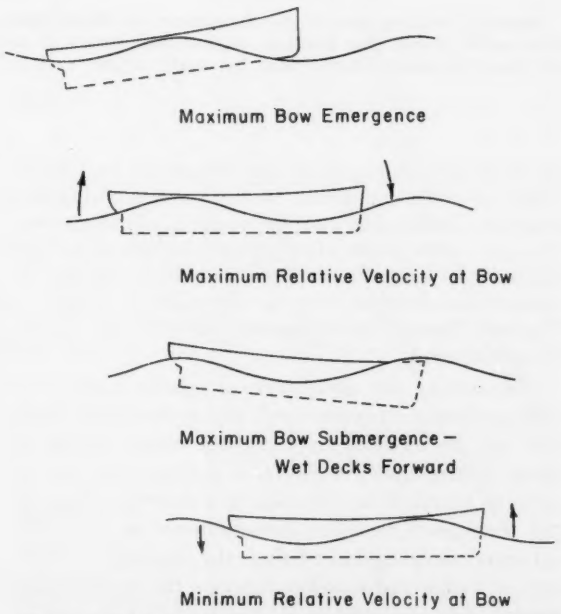

Figure 4. Typical Ship Behavior at Speeds for Synchronous Pitching in Regular Waves. This illustrates the characteristic "phase lag" of about 90 degrees between exciting moment and pitching response.

(2) phase relationships leading to wet decks and slamming also are characteristic of synchronism (see Figure 4), and (3) waves appreciably shorter than the length of a ship do not cause serious motions even at synchronism (see Figure 5). On the basis of

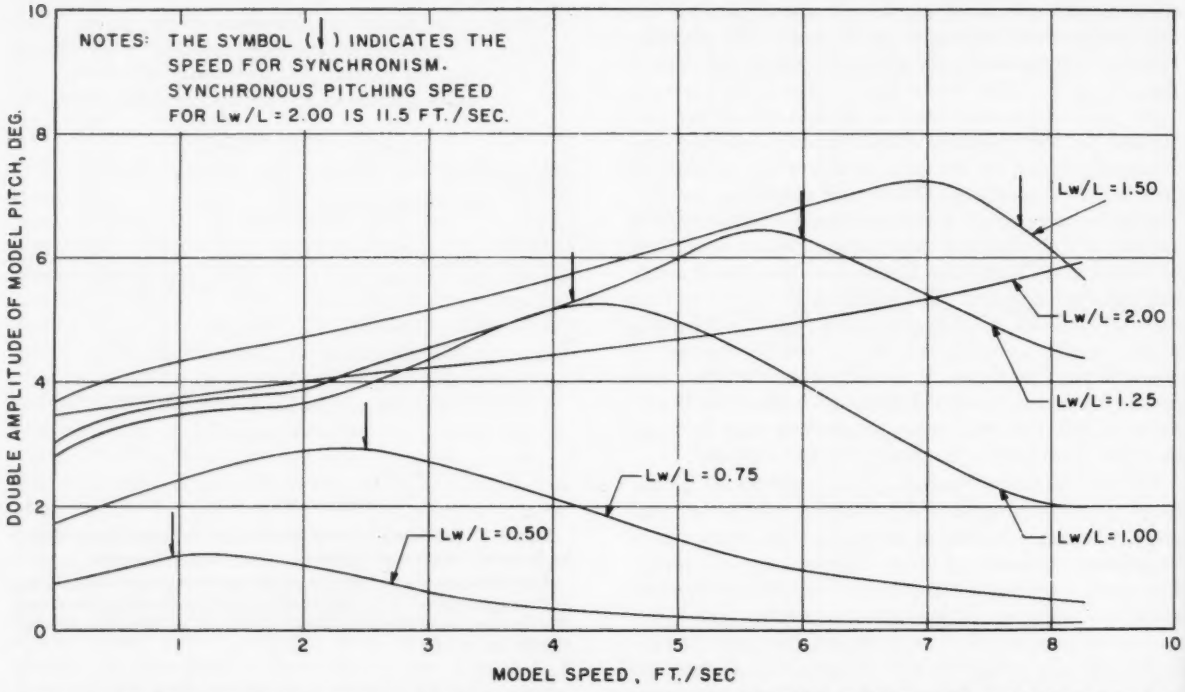

Figure 5. Pitching Amplitudes Determined by Model Tests of a Destroyer in Regular Head Seas. Wave height is equal to 1/48 model length. The locations of peak amplitudes of motion are indicated in relation to synchronism. Moderate amplitudes are shown in waves shorter than the model length, even at synchronism. From Lewis and Dalzell.[7]

these general facts, it appears there are two directions in which significant reduction of pitching and heaving amplitudes can be sought: avoiding synchronism with waves of ship length or longer, and reducing the magnification effect which causes increased amplitudes near synchronism as shown in Figure 3. Each of these possibilities will now be discussed in turn.

Considering the avoidance of synchronism, it is obvious that in a regular swell this undesirable condition can be avoided by changing either course or speed. If the speed is changed, synchronism can be avoided by either an increase or a decrease. Regarding the speed for pitch synchronism as the critical speed, slowing down brings the ship into the subcritical range and speeding up into the supercritical range.

The velocity of a single deep water wave is given by the "celerity"

$$c=\sqrt{gL_w/2\pi}$$

where L_w is the wave length, g is the acceleration of gravity, and any consistent units may be used (i.e., ft., lb., sec.). Although wave celerity increases with wave length, it turns out that the period of encounter with head seas increases as the waves become longer. This can be seen from the following equation for period of encounter:

$$T_e=\frac{L_w}{v+c}=\frac{L_w}{v+\sqrt{gL_w/2\pi}}$$

where v is the ship speed.

By equating the period of encounter, T_e, to the natural period of pitching, T_p, the critical speed, v_c, can be determined for any wave length. In nondimensional form it is given by

$$\frac{v_c}{\sqrt{gL}}=\frac{L_w/L}{T_p/\sqrt{L/g}}-\sqrt{(L_w/L)/2\pi}.$$

Or, with v_c in knots, L_w in feet and T_p in seconds,

$$\frac{v_c}{\sqrt{L}}=\frac{0.59\ (L_w/L)}{T_p/\sqrt{L}}-1.34\sqrt{L_w/L}.$$

This relationship is plotted in Figure 6 from Lewis.[3] For other headings a simple cosine correction must be introduced. The above expressions defining the conditions for synchronous pitching show two things. First, the critical speed is higher the longer the wave length. Second, the critical speed in any wave length is higher as $T_p/\sqrt{L}$ is reduced.

When a ship encounters irregular storm seas, the situation does not remain so simple. Oceanographers

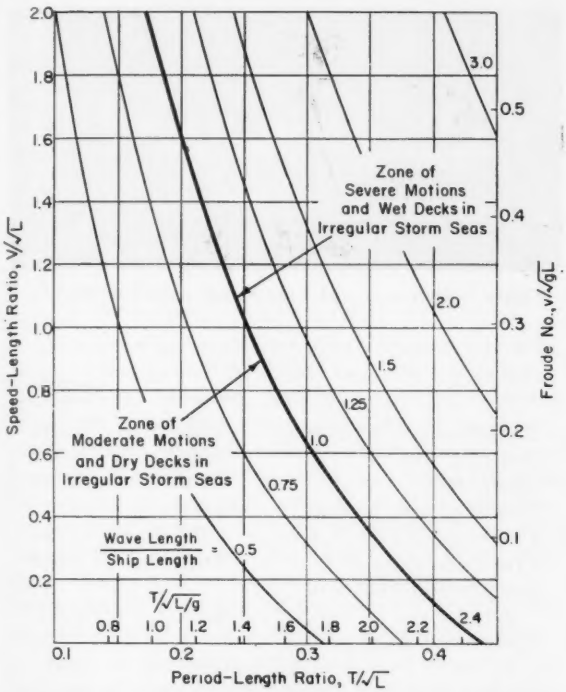

Figure 6. Theoretical Speed-Length Ratios for Synchronous Oscillation in Regular Head Seas of Different Lengths. Defining typical zones of severe and moderate motions in irregular seas.

have shown that storm seas can be considered as being composed of a great many regular wave trains, of varying length and direction of travel, all superimposed on one another.[8] They also have confirmed the observations of seamen that shorter waves are formed first in a storm. As the wind continues to blow, longer components are formed without seriously affecting the smaller components. For a particular wind velocity the sea reaches a limit when it attains its "fully developed" state. If the wind increases in strength, not only are the component waves believed to be higher, but—when the fully developed state is reached—longer components also will be present.

If a ship is able to attain a sufficiently high speed so that its period of encounter with the longest important wave component is shorter than the natural pitching period, the ship will be in the supercritical condition for that particular storm. Most ships can attain this condition only in moderately heavy seas, i.e., in light winds or in stronger winds of short duration. Hence, in general, whether or not a ship can attain the supercritical condition depends both on the sea state, as indicated by wind velocity and duration, and on the ship's natural pitching period. This is shown for the case of head seas in Figure 7 which is taken from Reference 3. At other headings the ship must reach even higher speeds to attain the supercritical condition.

Since short wave components are present in both severe and moderate storm seas, it is impossible to avoid synchronism with all component waves by reducing speed. However, model tests have shown that waves appreciably shorter than the ship do not cause serious motions even at synchronism (see Figure 5). Hence, speeds which are low enough to avoid synchronism with waves of ship length and longer generally lead to moderate motions. This condition may be termed the subcritical range, and for head seas it depends only on the ship length and the natural pitching period, i.e., on the ratio $T_p/\sqrt{L}$ as shown in Figure 6. All ships must at times reduce speed in storm seas to attain the subcritical condition. For seas approaching off the bow instead of from directly ahead, still lower speeds are required to attain the subcritical condition. The reason for this is that the effective wave-length increases as the heading angle to the waves increases. However, wave steepness is reduced on an oblique heading, and, therefore, the net

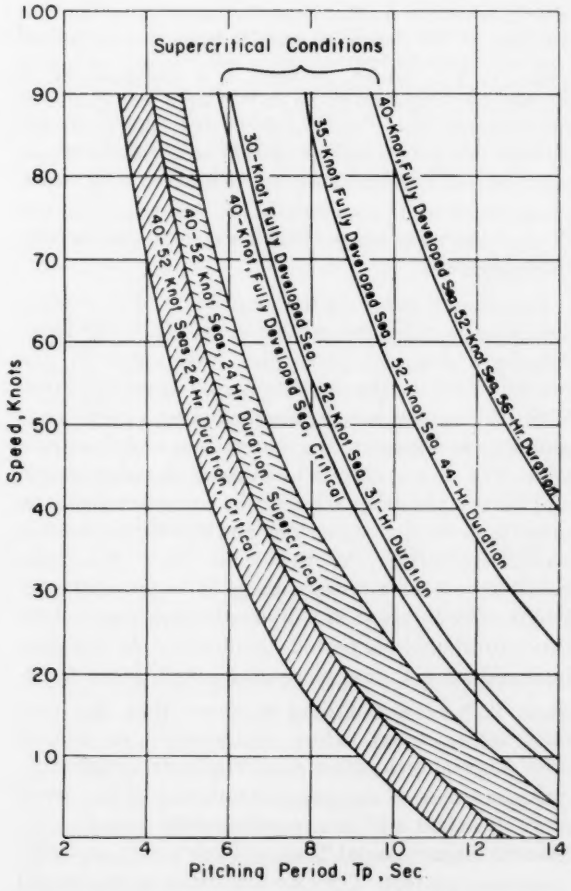
Supercritical Conditions
Speed, Knots
Pitching Period, Tp, Sec
40-Knot, Fully Developed Sea, 52-Knot Sea, 56-Hr Duration
35-Knot, Fully Developed Sea, 52-Knot Sea, 44-Hr Duration
30-Knot, Fully Developed Sea, 52-Knot Sea, 31-Hr Duration
40-Knot, Fully Developed Sea, Critical
40-52 Knot Seas, 24 Hr Duration, Supercritical
40-52 Knot Seas, 24 Hr Duration, Critical

Figure 7. Ship Speeds for "Critical" and "Supercritical" Operation.

result may be an improvement. In following seas most ships steaming at ordinary speeds are always in the subcritical range. This explains the advantage of heaving-to with wind and sea astern, provided that the ship can be kept under control.

A HIGH-SPEED SUPERCRITICAL SHIP

It may be of interest to consider possible ship designs aimed specifically at subcritical or supercritical operation.

Destroyers are much-maligned ships. They are small, their calm-water speed is high, and their rolling characteristics are bad. Hence, they are ideal candidates for the installation of anti-rolling fins. However, it perhaps will be agreed that destroyers can safely maintain much higher sea speeds than other vessels of their size—for example, small cargo ships. Violent motions in a rough head sea would cause both vessels to reduce power in order to ease slamming, to reduce shipping of water, or to cut down accelerations, but the safe speed of the destroyer would be considerably higher than the cargo ship. Likewise, at the same speed in the same head sea the destroyer would have less motion than the cargo ship. This advantage of the destroyer results from its low period ratio, $T_p/\sqrt{L}$, which, in turn, is a consequence of its slender proportions, as shown by a low displacement-length ratio, $\Delta/(L/100)^3$. In short, a destroyer can attain higher speeds in irregular storm seas before encountering synchronism with wave components of its own length, i.e., before getting out of the subcritical range. This is explained more fully in Reference 3.

One way of reducing the amplitudes of motions is therefore to raise the critical speed. In order to do this for a ship of a given size (displacement) one must first reduce the pitching period as much as possible. This can be done by concentrating weights near midships and keeping the waterplane as full as possible. The second step is to increase the ship length and reduce the other dimensions correspondingly in order to make the ship as long and slender as possible (low $\Delta/(L/100)^3$). At the same time the freeboard should be increased roughly in proportion to length. The increase in ship length also may reduce the natural pitching period significantly. At any rate, both reducing T_p and increasing length serve to reduce $T_p/\sqrt{L}$; and Figure 6 shows that this permits higher speed before synchronism is reached with wave components of ship length and longer.

An example of one possible outcome of this trend is a "stretched out" destroyer recently tested at the Stevens Experimental Towing Tank under an ONR-sponsored project. The characteristics of this model in comparison with a typical destroyer are given in Table I. Both models are shown in Figure 8.

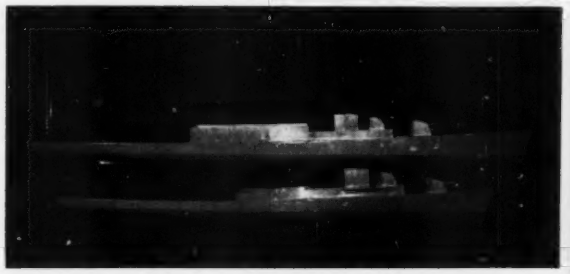

Figure 8. Destroyer and Lengthened Destroyer Models.

TABLE I

	Destroyer	Lengthened Destroyer
Length on LWL, L, feet	5.720	7.200
Breadth, feet	0.608	0.535
Draft, feet	0.208	0.186
Displacement, pounds	24.5	24.5
Freeboard at bow, F, feet	0.280	0.350
Freeboard ratio, F/L	0.049	0.049
Longitudinal radius of gyration	0.24L	0.24L
Natural pitching period, T_p, sec.	0.60	0.55
$T_p/\sqrt{L}$	0.25	0.205

It is believed that heavy weights such as fuel bunkers could be concentrated more amidships than is usually done, provided special attention is given to the matter in design. This would be of further advantage in reducing $T_p/\sqrt{L}$ for the lengthened ship. Admittedly there would be many serious problems in the design of such a slender ship, particularly longitudinal strength and transverse stability. It is assumed that anti-rolling fins certainly would be installed and would tend to minimize the problem of stability under way.

Ship motion studies suggest that further reduction in motions could be obtained by the use of more V-form sections forward, which would not only increase the damping but would tend to reduce the natural pitching period. Since destroyers have broad flat sterns, V-form forward sections also would provide better balance between the ends. Some increases in resistance could be accepted because the lengthened destroyer has significantly less ehp in calm water than the original design (about 40 percent less at 34 knots).

In order to compare the sea performance of these two ships, data obtained from model tests at the Stevens ETT in the same irregular head seas have been tabulated by Lewis and Numata[4] as shown in Table II. This sea pattern corresponded closely to that expected at sea when a 32-knot wind has blown steadily for about 18 hours. The table shows that under comparable conditions the motions of the lengthened ship are appreciably less than the conventional de-

stroyer. The accelerations also were reduced correspondingly.

TABLE II

Ship Speed, Knots	Ship	Dbl. Pitch Amplitude, Deg. Average	Dbl. Pitch Amplitude, Deg. Average Highest 10%	Dbl. Heave Amplitude, In. Average	Dbl. Heave Amplitude, In. Average Highest 10%
0	Destroyer	5.3	10.8	1.12	2.28
0	Long Destroyer	3.5	6.5	0.65	1.39
12	Destroyer	6.7	13.7	1.64	3.33
12	Long Destroyer	4.1	7.3	0.82	1.71

A study of comparative tests in regular waves by Numata and Lewis[4] indicates that in a storm sea in which the conventional destroyer can attain a speed of say 20 knots, the longer vessel should be able to attain at least 30 knots with equal amplitudes of motions, which is a 50 percent increase in speed. It should be recognized, however, that the vertical accelerations would be greater in the latter case because of the shorter natural pitching period of the lengthened ship. (Maximum acceleration $=2\pi S^2/T$, where S is the amplitude of motion.) Consequently, the speed for equal accelerations is somewhat lower than 30 knots.

It will be noted that the freeboard ratio for the lengthened destroyer model was the same as the destroyer, and, as a result, the forward deck usually was dry in irregular tank waves. Because of the model's fineness, no difficulty in respect to slamming was encountered. In short, the long slender ship offers distinct possibilities for reduced motions in rough seas and, hence, attaining higher sea speed. However, there is obviously a practical limit on how long and slender a ship can be built.

A HIGH-SPEED SUPERCRITICAL SHIP

Another approach to the problem of reduced motions and higher speeds in rough seas is to strive for supercritical operation. This is a difficult condition to achieve except in moderate seas, but the possibility is being explored at the Stevens ETT. The basic aim is to attain a long natural pitching period and high ship speed. The latter requirement means that a slender ship like a destroyer is needed for low resistance, but one not necessarily as extreme as the lengthened subcritical ship described above. Arrangement of ballast tanks and other heavy weights near the ends would help to lengthen the pitching period, but a very fine waterline would be even more helpful. In view of the fact that a ship in supercritical operation tends to plunge through the waves rather than to ride over them, it would seem to be infeasible to attempt to keep water off the foredeck. Accordingly, a low freeboard and a heavily built deck would seem to be the best way to cope with the water and at the same time would permit a very narrow waterline. In short, an ideal ship would be a surfaced submarine made longer and more slender in order to attain high surface speed.

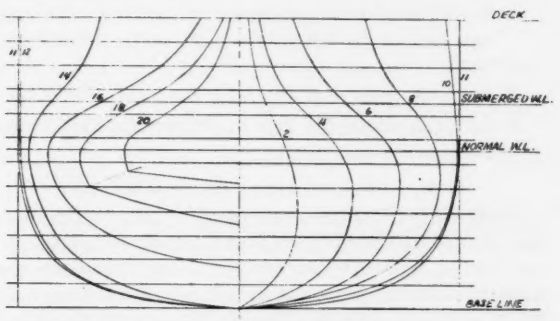

Figure 9. Body Plan of Proposed Ship for High-Speed Supercritical Operation.

Figure 9 illustrates a possible design for a supercritical ship of this type. The normal waterline would be used for good weather operation, while in bad weather large peak ballast tanks would be filled for the purpose of lengthening the pitching period in several ways: by increasing the displacement, by increasing the radius of gyration, and by reducing the waterplane area. Anti-rolling fins also would be an important feature of this design. A model is being built to the lines of Figure 9. Characteristics of the model in comparison with the typical destroyer are shown in Table III. It is estimated that a 400-foot ship with an eight second pitching period might be expected to attain the supercritical condition in most storms at a speed of 40 to 45 knots (see Figure 7).

TABLE III

	Destroyer	Semi-submerged Ship*
Length of LWL, feet	5.720	5.720
Breadth, feet	0.608	0.540
Draft, feet	0.208	0.250
Displacement, pounds	24.5	27.7
Freeboard at bow, F, feet	0.280	0.100
Freeboard ratio, F/L	0.049	0.017
Longitudinal radius of gyration	0.24 L	0.26 L
Natural pitching period, T_p, sec.	0.60	0.92
$T_p/\sqrt{L}$	0.25	0.38

*At deep draft for high speed operation in rough seas.

Many technical problems would be involved in the practical design of such a supercritical ship: very large power, sufficient interior space with a small deckhouse and large ballast tanks, obtaining an adequate structure to withstand high impact loads, etc. Undoubtedly the problems could be solved if there were a military mission for a vessel of the following characteristics:

1. Very high speed in both calm and rough seas,

2. Small amplitudes of motions (rolling, pitching, and heaving), but fairly high vertical accelerations,
3. Possibly high impact and vibratory oscillations,
4. Limited space for crew accommodation, armament and equipment.

SHIP IMPROVEMENTS IN CRITICAL RANGE

Practically all ships when heading into rough head seas at normal speeds soon find themselves in the critical range where synchronism occurs between the natural pitching period and a band of wave components of ship length and longer. Violent pitching and heaving motions, with high vertical accelerations, are then experienced—along with wet decks and sometimes slamming. Slowing down, or turning and running away from the seas, will invariably bring the ship into the subcritical range where in synchronism occurs only with component waves less than ship length. Motions are then more moderate and phase relationships are such that less water is shipped forward and slamming, if any, becomes less frequent. This raises the question as to what means can be applied in the design of a ship to reduce the amplitudes of motions when the ship attempts to operate in the critical range, assuming that the main ship characteristics and proportions have been settled.

If one is concerned mainly with the shipping of water and slamming, much can be gained by increasing freeboard, improving flare, reducing bottom flatness, etc. but for the reduction of motions, which is the subject of this paper, more drastic measures must be considered. It has been shown that the most violent pitching motion is generally in response to the near-synchronous wave components. Under such circumstances, as with any oscillating system, simple damping is an effective way to reduce motion amplitudes (see Figure 3). Two possible ways of doing this appear: modification of the hull form and installation of auxiliary devices.

Theory and experiment have shown that filling out the waterline is a hull form modification which is beneficial in reducing pitching and heaving. Hence, any increase of beam and use of more V-form sections are generally desirable steps in the design stage. Fortunately these measures also tend to shorten the natural pitching period, which raises the critical speed range slightly. Abkowitz[5] has shown that concentrating weights near midships increases the damping, as well as shortening the pitching period. Important as these effects are, the writer does not expect more than modest reductions in motion amplitudes by such means.

The use of external devices offers somewhat more promise than changing the hull form. The evaluation of different possibilities requires a consideration of the characteristic phase relationships between the motions at the ends of the ship and the encountered waves at synchronism. A study along this line at the Stevens ETT showed that the typical flow at the bow

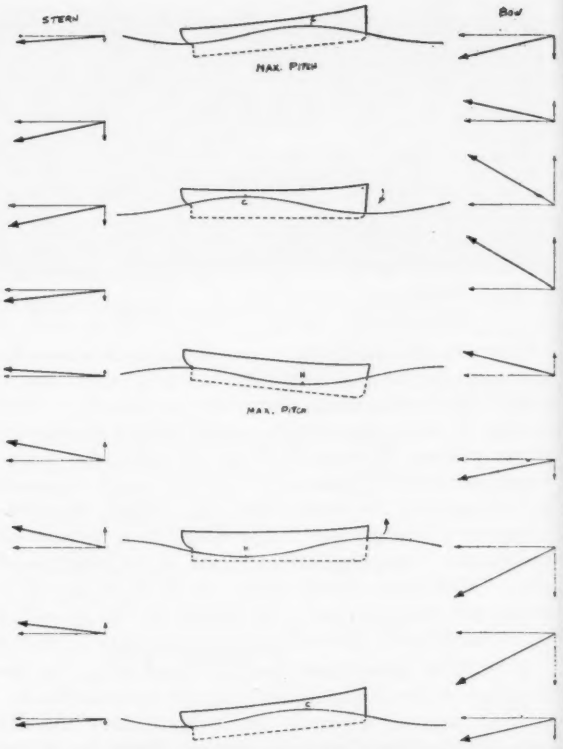

Figure 10. Diagrams Showing Direction of Flow at Bow and Stern. Series 60 ($C_b = 0.60$) Model in Waves 1½ x Model Length, Height = 1/48 Model Length at Synchronous Speed (No Fins).

has a wide variation in direction during a motion cycle, so that fixed fins in this location develop large angles of attack. At the stern the direction changes are much smaller (see Figure 10). This means that fixed fins should be very effective devices near the bow but not at the stern, a fact which experiments have confirmed. On the other hand, theory and experiment indicate good potentialities for controllable stern fins which can develop high lift by adjustment of their angle of attack. Table IV gives the amplitudes of motion for a Mariner model in waves of 1.3 x model length and height 1/60 of the length at a ship speed of 15 knots with different fin arrangements:

TABLE IV

	Double Amplitude, Degrees
No fins	7.4
Fixed bow fins	6.4
Fixed bow and stern fins	6.0
Fixed bow fins and movable stern fins	5.4

In the case of the movable stern fins, a sinusoidal motion was given by means of a motor, its phase being adjusted to give the greatest possible reduction in

motion amplitudes. For such a device to be effective in irregular storm sea conditions an elaborate control system, of course, would be required.

Results of trials of fixed bow fins on the U.S.S. Compass Island, a converted Mariner, will be awaited with great interest.

CONCLUSIONS

Although the problem of reducing ship motions at sea is difficult, several directions for obtaining this end are available. Simple improvements in hull form or shape seem to be less promising than radical changes in proportions and other basic characteristics. These radical developments involve obtaining either sub or supercritical operation, if possible. In addition, fixed bow fins and/or controllable stern fins appear promising for ships which at times must operate in the critical pitching range, while anti-rolling fins already have brought the problem of roll and side sway near solution.

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The detection and measurement of vibration has received increasing emphasis in engineering applications in recent years. Three basic instruments used to determine vibration magnitudes are the displacement pickup, the velocity pickup, and the acceleration pickup. The displacement, or seismic type, operates on the principle that a spring suspended mass in a supporting case, properly damped, will remain essentially motionless as the case is subjected to forced vibration, provided that the forcing frequency is sufficiently greater than the natural frequency of the instrument. A second type uses a permanent magnet as a suspended mass, which is surrounded by a coil attached to the case. Relative motion between the coil and case generates a voltage which is proportional to the velocity of displacement. When the frequency of the vibration to be measured is relatively low, up to about one-half the natural frequency of the suspended system, the relative displacement between the mass and the case is proportional to the acceleration. Various ingenious schemes are used to sense and display the acceleration. Variations of these three basic instruments are used for special purposes, such as for measuring torsional vibration.

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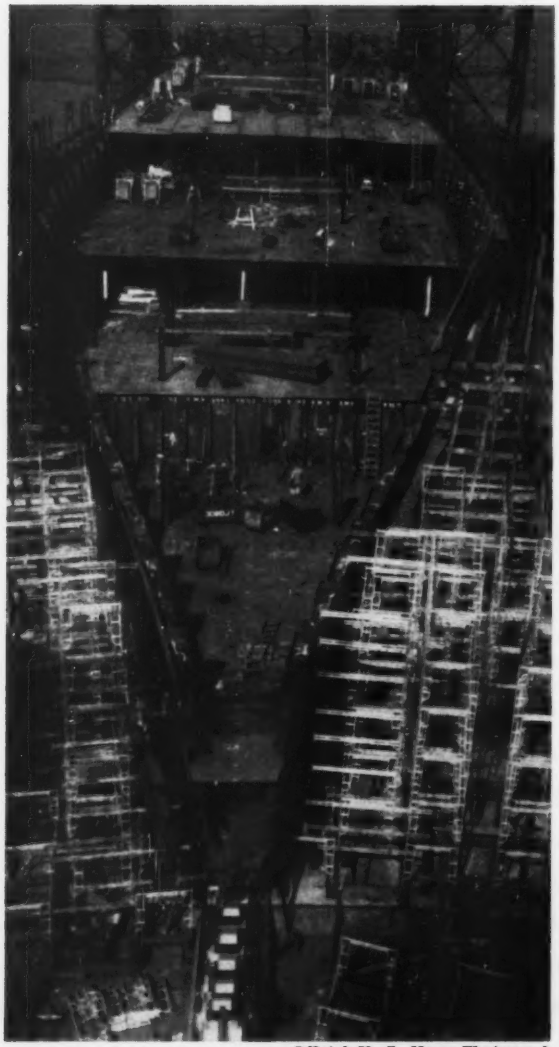
September 1958

One of the Navy's Most Modern Underway Fleet Replenishment ships takes shape on the ways.



Official U. S. Navy Photograph

A view of the second of three modern ammunition ships, PYRO, as she rapidly approaches the point in erection when she will be launched at the Bethlehem-Sparrows Point Shipyard. It is to be noted that the forward portions of the main deck, second deck, and first platform are yet to be fitted and installed prior to installation of the prefabricated bow section.

Official U. S. Navy Photograph

A photograph of the same ammunition ship, taken one month later. The ship is now being prepared for installation of the prefabricated bow section and the riveted main deck gunwale angle preparatory to launching.

TETSUO NISHIYAMA

HYDRODYNAMIC INVESTIGATION OF THE SUBMERGED HYDROFOIL

THE AUTHOR

studied Naval Architecture mainly at the department of Naval Architecture of Tokyo University, Tokyo, Japan from 1944-1947. After several post-graduate courses, he has become an assistant professor in the department of Mechanical Engineering of Tohoku University in Sendai. His work is connected with general hydromechanical problems in Naval Architecture, in particular with wave resistance theory and with the submerged hydrofoil.

PART II

A method for obtaining the profile form of the submerged hydrofoil with a prescribed pressure distribution.

SYMBOLS

C_a = lift coefficient $(L/\frac{1}{2}\rho C^2 l)$
C_w = wave-making resistance coefficient $(R/\frac{1}{2}\rho C^2 l)$
l = chord length
f = immersion depth, which is defined as a vertical distance from still water level to the mid-point of chord length
α = angle of attack
g = gravitational acceleration
K_0 = g/C^2
Subscript $_0$ means that the matter is not under the effect of the free water surface
Subscript $_e$ means that the matter is under the effect of the free water surface

INTRODUCTION

The pressure distribution is closely related to the transition and separation of the boundary layer, and the inception of cavitation over the back of a hydrofoil. Therefore, in order to research the suitable profile form of the hydrofoil for a given condition of operation, it is of primary importance to establish a theoretical method for obtaining the profile form of the hydrofoil with a prescribed pressure distribution under a given condition.

From the standpoint of thin wing theory, Warren [1] proposed a method for obtaining the profile form, in particular delaying the occurrence of cavitation. However, this method neglected the existence of the free water surface which has a unique and important effect.

Generally the existence of the free water surface decreases the lift of the submerged hydrofoil for normal speeds. Hence, if we apply the profile form designed by Warren's method to the submerged hydrofoil, we meet inevitably with the fact that the lift coefficient of the submerged hydrofoil is smaller than the designed lift coefficient, even if the occurrence of cavitation might have been delayed. This shows that Warren's method is on the dangerous side.

Under these circumstances, taking the existence of the free water surface into account, a method for

obtaining the profile form of the submerged hydrofoil with a prescribed pressure distribution is proposed from the standpoint of the theory of the submerged hydrofoil of arbitrary section.[2] At the same time the suitability of the types of the pressure distribution is discussed for the submerged hydrofoil.

BASIC IDEA

As the pressure distribution can be easily converted into the velocity distribution from Bernoulli's theorem, we hereinunder use only the latter.

It is rather difficult to treat the prescribed velocity distribution in that condition under the effect of the free water surface; therefore, after converting the prescribed velocity distribution into the corresponding distribution in an infinite fluid (without the effect of the free water surface), we calculate the profile form of the submerged hydrofoil from that corresponding distribution by the already known method.[3]

If the velocity distribution in the case where a hydrofoil is affected by the free water surface and in the other case where it is not affected by the same (i.e. in an infinite fluid) are denoted by v_e and v_o respectively, and the quantity of the change in the velocity distribution due to the effect of the free water surface is denoted by Δv_e, we have the relation

$$v_e/C=v_0/C+\Delta v_e/C \quad \ldots\ldots (1)$$

where C is the advancing speed of the hydrofoil. Since the lefthand side of (1) is given, if the second term of the righthand side can be obtained, it is possible to convert it into the velocity distribution in an infinite fluid, and consequently to calculate the profile form of the hydrofoil by the known method.[3] At the first stage of calculation, $\Delta v_e/C$ cannot be obtained directly since the profile form of the hydrofoil is still not known. However it can be obtained by successive approximations.

FIRST APPROXIMATION

For the first stage of approximation we assume that $\Delta v_e/C$ is equal to that of the plate section. Then we can use the theoretical solution [4] for the hydrofoil of the plate section.

For a hydrofoil of the plate section not affected by the free water surface:

Lift coefficient: $$C_{y_0}=2\pi\sin\alpha_0 \quad \ldots\ldots (2)$$

Velocity distribution: $$v_0/C=\cos\alpha_0+\cot\frac{\eta}{2}\sin\alpha_0 \quad \ldots\ldots (3)$$

Coordinate: $$x/l=\tfrac{1}{2}\cos\eta;\ 2\pi\geqq\eta\geqq 0 \quad \ldots\ldots (4)$$

For a hydrofoil of the plate section affected by the free water surface:

Lift coefficient:
$$C_{y_e}=2\pi\sin\alpha_e[k-\tfrac{1}{2}K_0 l\sin\alpha_0\{k^2r_0+\tfrac{1}{2}K_0 l k r_1\sin\alpha_e+\tfrac{1}{4}K_0^2l^2(\tfrac{1}{2}k^2 s\sin 2\alpha_e-\tfrac{1}{4}r_2-k^2r_0r_1\sin^2\alpha_e-k^2r_1 s\sin\alpha_e\cos\alpha_e)\}+\cdots] \quad \ldots\ldots (5)$$

where $$k=1+\tfrac{1}{2}a_1(K_0l)+\tfrac{1}{4}a_2(K_0l)^2+\cdots \quad \ldots\ldots (6)$$

with
$$a_1=-(r_0\sin\alpha_e+s\cos\alpha_e)$$
$$a_2=-(r_0\sin\alpha_e+s\cos\alpha_e)^2-\tfrac{1}{2}r_1(1+\cos 2\alpha_e)+\tfrac{1}{2}s\sin 2\alpha_e$$

and $$s=2\pi e^{-2K_0 f}$$

$$r_n=\frac{n!}{(2K_0f)^{n+1}}+2\left\{\frac{(n-1)!}{(2K_0f)^n}+\cdots+\frac{1}{2K_0f}-e^{-2K_0f}\mathrm{Ci}(e^{2K_0f})\right\} \quad \ldots\ldots (7)$$

Velocity distribution:

$$v_e/C=\frac{2\sum_1^\infty(C_{n\cdot r}\sin n\eta+C_{n\cdot i}\cos n\eta)+k}{\sin\eta}\sin\alpha_e \quad \ldots\ldots (8)$$

where
$$C_{1\cdot r}=\tfrac{1}{2}\cot\alpha_e-\tfrac{1}{4}kK_0l(r_0\cos\alpha_e-s\sin\alpha_e)+\frac{1}{16}K_0^2l^2s+\cdots$$

$$C_{2\cdot r}=-\frac{1}{16}k\,K_0^2l^2(r_1\sin 2\alpha_e+s\cos 2\alpha_e)+\cdots$$

$$C_{1\cdot i}=\tfrac{1}{2}-\tfrac{1}{4}k\,K_0l(r_0\sin\alpha_e+s\cos 2\alpha_e)+\frac{1}{16}K_0^2l^2r_1+\cdots$$

$$C_{2\cdot i}=\frac{1}{16}kK_0^2l^2(r_1\cos 2\alpha_e-s\cos 2\alpha_e)+\cdots$$

In general there is the relation

$$C_a=\frac{A-C_w\sin\alpha}{\cos\alpha} \quad \ldots\ldots (9)$$

between an area A surrounded by the pressure distribution curve in the usual non-dimensional form in wing theory, and the lift coefficient C_a and wave-making resistance coefficient C_w. Since the angle of attack is small, it is given with sufficient accuracy by

$$C_a=A \quad \ldots\ldots (9')$$

When the immersion depth and the speed of the hydrofoil are given, α_0 and α_e are obtained from (2), (5) and (9′), respectively. Substituting α_0 and α_e into (3) and (8), respectively, we may obtain $\Delta v_e/C$. Then v_0/C can be obtained from (1), and consequently the profile form of the hydrofoil can be calculated.

SECOND APPROXIMATION

In order to increase the accuracy of calculation, it is necessary to calculate $\Delta v_e/C$ for the profile form of the hydrofoil obtained by the first approximation. Then the theoretical solution [2] for the submerged hydrofoil of an arbitrary section can be used for this purpose.

For a hydrofoil of an arbitrary section not affected by the free water surface:

Lift coefficient: $$C_{a_0}=8\pi k_0\sin(\alpha_0+\beta_0) \quad \ldots\ldots (10)$$

Velocity distribution:

$$v_0/C=\frac{2k_0\{\sin(\alpha_0+\beta_0)+\sin(\alpha_0+\beta_0-\eta)\}}{\{\tfrac{1}{4}\sin^2\eta+(\sum_1^\infty na_n\sin n\eta+\sum_1^\infty nb_n\cos n\eta)^2\}^{\frac{1}{2}}} \quad \ldots\ldots (11)$$

where

$$k_0=\tfrac{1}{2}\{(\sum_1^\infty na_n)^2+(\tfrac{1}{2}+\sum_1^\infty nb_n)^2\}^{\frac{1}{2}},$$

$$\beta_0=\tan^{-1}\left|\sum_1^\infty na_n/(\tfrac{1}{2}+\sum_1^\infty nb_n)\right|$$

Coordinate: $$x/l=\tfrac{1}{2}\cos\eta,\ y/l=\sum_0^\infty a_n\cos n\eta-\sum_1^\infty b_n\sin n\eta; \quad \ldots\ldots (12)$$
$$0\geqq\eta\geqq-2\pi$$

For a hydrofoil of an arbitrary section affected by the free water surface:

Lift coefficient:
$$C_{a_e}=8\pi k_0 k\sin(\alpha_e+\beta_0)-4\pi k_0^2\sin^2(\alpha_e+\beta_0)\,[4k^2K_0lr_0$$
$$+8k^2K_0^2l^2r_1k_1\sin(\alpha_e+\beta_0)-4kK_0^2l^2r_1\{k_2\cos(2\alpha_e+\beta_0+\beta_2)-k_0\}\times\frac{1}{\sin(\alpha_e+\beta_0)}+\text{- - -}] \quad \ldots\ldots (13)$$

where

$$k=1+a_1(K_0l)+a_2(K_0l)^2+\text{- - -} \quad \ldots\ldots (14)$$

with

$$a_1=-2k_0\{r_0\sin(\alpha_e+\beta_0)+s\cos(\alpha_e+\beta_0)\}$$
$$a_2=-4k_0^2\{r_0\sin(\alpha_e+\beta_0)+s\cos(\alpha_e+\beta_0)\}^2$$
$$-2k_0k_1\{r_1\cos(\beta_1-\beta_0)-s\sin(\beta_1-\beta_0\}$$
$$+2r_1\{k_0k_1\cos(\beta_0+\beta_1+2\alpha_e)-k_0^2\cos(2\alpha_e+2\beta_0)\}$$
$$-2s\{k_0k_1\sin(\beta_0+\beta_1+2\alpha_e)-k_0^2\sin(2\alpha_e+2\beta_0)\}$$
$$+r_1\{k_2\sin(\alpha_e+\beta_2)+k_0\sin(\alpha_e+\beta_0)\}\times\frac{k_0}{\sin(\alpha_e+\beta_0)}$$
$$-s\{k_2\cos(\alpha_e+\beta_2)-k_0\cos(\alpha_e+\beta_0)\}\times\frac{k_0}{\sin(\alpha_e+\beta_0)}$$

and

$$k_1 e^{i\beta_1} = a_0$$

$$k_2 e^{i\beta_2} = -b_1 + \tfrac{1}{2}(\tfrac{1}{2} + \sum_1^\infty n b_n) + i(a_1 - \tfrac{1}{2}\sum_1^\infty n a_n)$$

Velocity distribution:

$$v_e/C = \frac{I}{\left\{\tfrac{1}{4}\sin^2\eta + \left(\sum_1^\infty na_n \sin n\eta + \sum_1^\infty nb_n \cos n\eta\right)^2\right\}^{\frac{1}{2}}} \quad \cdots\cdots\cdots\cdots (15)$$

where

$$I/2k_0\sin(\alpha_e+\beta_0) = k + k_0\sin(\alpha_e+\beta_0-\eta)\left[\frac{1}{k_0\sin(\alpha_e+\beta_0)}\right.$$
$$-2K_0 lkr_0 - K_0^2 l^2 (b_{00.1}{}^* r_1 - b_{00.2}{}^* s) - 2kk_1 K_0^2 l^2 \{r_1 \sin(\alpha_e+\beta_1)$$
$$+ s\cos(\alpha_e+\beta_1)\} - 2kk_0 K_0^2 l^2 \{r_1 \sin(\alpha_e+\beta_0-\eta)$$
$$+ s\cos(\alpha_e+\beta_0-\eta)\} + \text{- - - }] + k_0\cos(\alpha_e+\beta_0-\eta)\,[2K_0 lks$$
$$+K_0^2 l^2 (b_{00.1}{}^* s + b_{00.2}{}^* r_1) - 2kk_1 K_0^2 l^2 \{r_1\cos(\alpha_e+\beta_1)$$
$$-s\sin(\alpha_e+\beta_1)\} - 2kk_0 K_0^2 l^2 \{r_1\cos(\alpha_e+\beta_0-\eta)$$
$$-s\sin(\alpha_e+\beta_0-\eta)\} + \text{- - - }]$$

and

$$b_{00.1}{}^* = -\{k_2\cos(2\alpha_e+\beta_0+\beta_2) - k_0\} x \frac{1}{\sin(\alpha_e+\beta_0)} + 2k_1\sin(\alpha_e+\beta_1)$$

$$b_{00.2}{}^* = k_2\sin(2\alpha_e+\beta_0+\beta_2) x \frac{1}{\sin(\alpha_e+\beta_0)} + 2k_1\sin(\alpha_e+\beta_1)$$

The coefficients a_n and b_n can be determined by carrying out the Fourier analysis for the profile form of the hydrofoil obtained by the first approximation. The said form in the second approximation can be calculated by the use of a_n and b_n and by the same procedure as that of the first approximation.

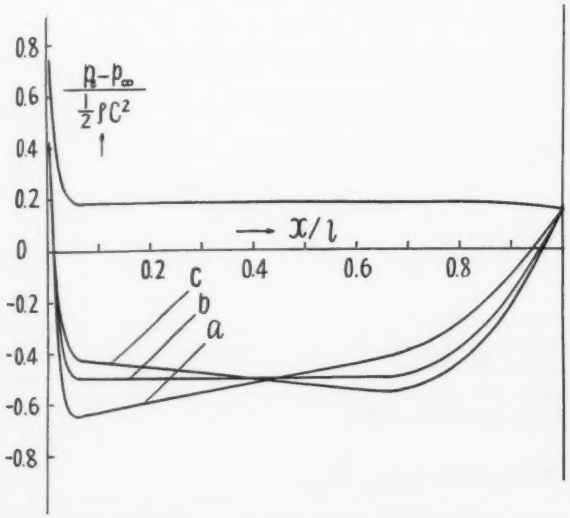


Figure 1. Three types of the prescribed pressure distribution with $C_{s_e}=0.6$.

NUMERICAL EXAMPLES

The pressure distribution, aimed at the lift coefficient $C_{s_e}=0.6$ is shown in Figure 1(a). It has a minimum immediately after the leading edge and a gradual rise to the trailing edge. From this pressure distribution the profile form was calculated under the condition:

Immersion depth $f/l=1.0$

Froude number $C/\sqrt{gl}=3.16$ $(K_0 l=0.1)$

and shown in Figure 2; the difference between the first and second approximation is that in the latter the thickness is greater in the fore part and smaller in the after part and the camber is smaller in the latter, but these amounts are very small.

Now we calculate the profile form ignoring the effect of the free water surface, as per Warren's method. From the pressure distribution shown in Figure 1(a), the profile form obtained is as indicated by the dotted line in Figure 2; the feature of which is that the thickness is smaller but the camber is almost constant.

When this profile form is used as that of the submerged hydrofoil in the above mentioned condition, the lift coefficient can be calculated by applying the theory [2] of the submerged hydrofoil of arbitrary section; we obtain the lift coefficient $C_{s_e}=0.4665$ which is approximately 20 percent less than the designated lift coefficient $C_{s_e}=0.6$. This decrement is exclusively due to the effect of the free water surface. Hence this fact tells the importance of considering the existence of the free water surface in the design of the profile form of a submerged hydrofoil.

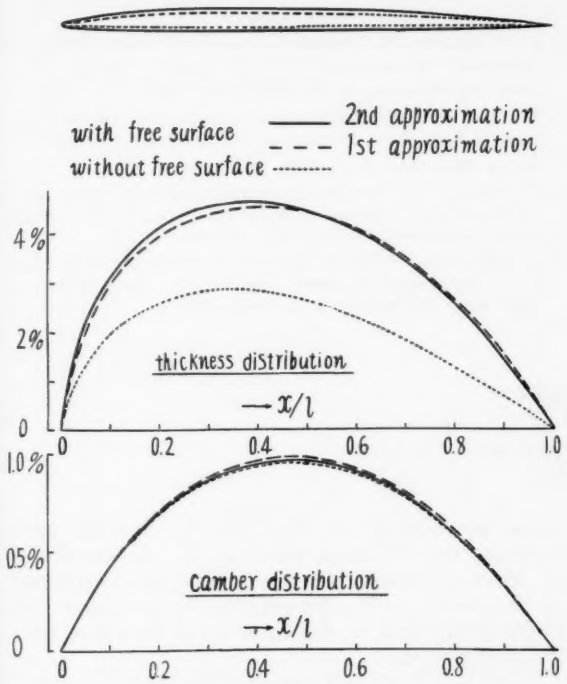


Figure 2. Comparison between first and second approximations of the profile form corresponding to the prescribed pressure distribution (a) in Figure 1
(K_0l=0.1 f/l=1.0)

SUITABILITY OF THE TYPES OF PRESSURE DISTRIBUTION

It is an important matter in the design of the profile form to examine the suitability of the types of the pressure distribution. When we choose the pressure distribution for designing the submerged hydrofoil, the following items should be taken into account:

1. make the minimum pressure as small as possible, in order to avoid the occurrence of cavitation.
2. make the pressure gradient immediately before the trailing edge as loose as possible, in order to decrease the form resistance due to the separation of the boundary layer.
3. make the position of the minimum pressure back toward the trailing edge, in order to decrease the frictional resistance.
4. make the wave disturbance on the water surface as small as possible, in order to decrease the wave-making resistance.

Now we examine the relation between the wave-making resistance and the types of pressure distribution. The three types of pressure distribution on the back with the designed lift coefficient C_{Le}=0.6

	a	b	c
C_w	0.0163	0.0163	0.0165

will be considered and these are shown in Figure 1 by (a) (pressure rise), (b) (pressure constant) and (c) (pressure descent). From these the profile forms are obtained under the condition

Immersion depth f/l=1.0
Froude number $C/\sqrt{gl}$=3.16 (K_0l=0.1)

and shown in Figure 3.

For these three profile forms the wave-making resistance coefficients can be calculated as follows: The difference between these three wave-making resistance coefficients is negligibly small; hence it is not too much to say that, so far as the wave-making resistance is concerned, the type of pressure distribution has little significance as long as the lift coefficient is maintained the same.

From the above results the types of pressure distribution should be selected, taking items [1] [2] [3] into account.

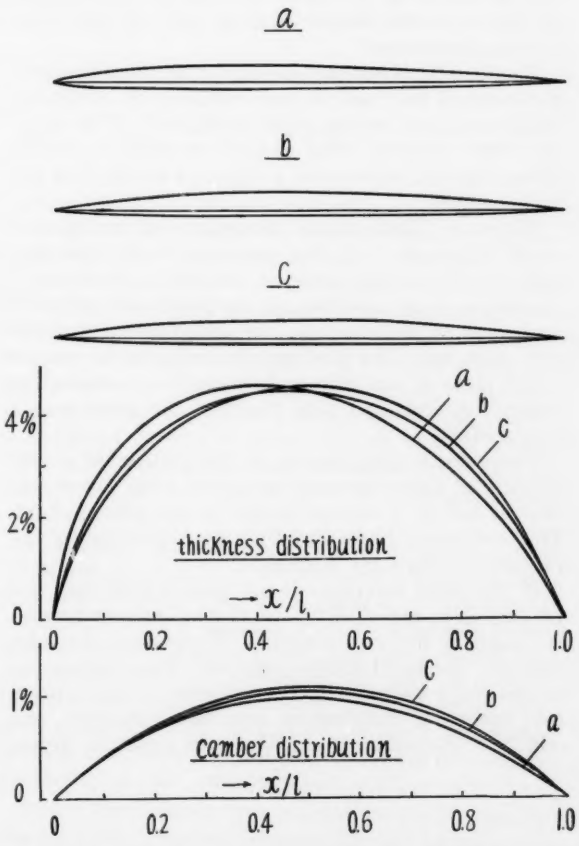


Figure 3. Profile form corresponding to the three prescribed pressure distributions (a) (b) and (c) in Figure 1
(K_0l=0.1 f/l=1.0)

CONCLUSIONS

The main problems treated in this paper are as follows:

1. Applying the theory of the submerged hydrofoil of an arbitrary section, a method for obtaining the profile form of the submerged hydrofoil with a prescribed pressure is presented.
2. It is shown that neglecting the effect of free water surface, such as Warren's method, leads to the dangerous side in the design of the profile form of the submerged hydrofoil.
3. The suitability of the types of pressure distribution is examined for the submerged hydrofoil.

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DISCUSSION
Review of
Professor Nishiyama's Paper on Hydrofoils
(Part II)

There are two closely-related problems in airfoil (or hydrofoil) theory. The first is the determination of the characteristics (lift, drag, moment, pressure distribution and flow pattern) of any selected foil. The second is the so-called inverse problem, where the pressure distribution is given and the foil shape is to be determined.

The first problem was solved for an arbitrary two-dimensional foil near a free surface, by Professor Nishiyama in a recent paper published in the August 1958 JOURNAL. The second problem is herein presented and represents a logical extension of the earlier study.

Professor Nishiyama's procedure is straightforward, although, as in his previous work, considerable developmental work is omitted. A method of successive approximations is outlined such that the accuracy of determining the foil shape improves with each step. The first approximation is to assume a flat plate in an infinite fluid and to compare its velocity distribution with that of a flat plate near a free surface.

The velocity distribution of the plate near a free surface is then assumed to arise from some unknown foil of a certain shape in an infinite fluid. This unknown shape may then be calculated by the procedure given by Warren.

If this latter reference is not readily available, the process is in many of the textbooks on aerodynamics, such as Milne-Thomson's "Theoretical Aerodynamics," Second Edition, page 148. This method involves the separation of the velocity, or its equivalent pressure, distribution into two functions, one odd and the other even. The odd function determines the camber line function, Y_c, and the even function, the thickness function, Y_t. As developed by Milne-Thomson, the profile can be determined by two relations, Y_u, the equation of the upper side of the profile and Y_l, the equation of the lower side. We then find that

$$Y_u = Y_c + Y_t \text{ and } Y_l = Y_c - Y_t.$$

Professor Nishiyama uses the same procedure for his second, and more accurate, approximation. Here he compares the velocity distribution of the profile, developed in the first approximation, in an infinite fluid and near a free surface. In this he utilizes the results of his earlier paper. Then the procedure for obtaining the profile is indicated to be the same as for the first approximation.

In the practical design of hydrofoils, it is necessary to determine the lift, drag and cavitation characteristics. The usual procedure is to select a profile from the many series available in the literature, such as presented in NACA Report No. 824. The profile is picked so as to have the desired characteristics. The more desirable approach is the one given by Professor Nishiyama in this paper. A pressure distribution can be quite readily selected that will yield the desired characteristics; the determination of the foil shape from this should yield a more satisfactory hydrofoil than one selected from an arbitrary series of foils. It should be noted, however, that an extension of this paper to foils of finite aspect ratio would provide information much nearer the needs of the designer.

In this, and in the earlier paper, Professor Nishiyama develops tools and methods that should eventually prove quite useful to the hydrofoil designer.

"ENGINEERING"

NUCLEAR REACTOR FOR DISTILLING SEA-WATER

ACKNOWLEDGEMENT

The authors of this article, published in the 16 May 1958 edition of "Engineering," were I. Vilentchuk and N. Arad, of the Mekoroth Water Company, Tel Aviv, Israel.

SUMMARY

THE OBJECT of the present article is to show that a pressurized-water nuclear reactor producing low temperature steam for distillation of sea-water for irrigation, domestic and industrial usage, will operate in highly favorable conditions with a steady load during practically 8,760 hours per year. It is maintained that the constructional and technological features of a pressurized-water reactor used for generating low temperature steam will be much simpler than for one working in conjunction with an electric power plant. Moreover, certain nuclear features of the Low Temperature Reactor (LTR) may be favorably compared with those of the Shippingport type reactor. By analyzing cost components of heat generated by an atomic power plant and those of a Low Temperature Reactor working in conjunction with a sea-water demineralization plant, the total cost of distilling water is obtained.

The greatest effort of engineering thought is directed towards the use of heat extracted from nuclear reactors for the generation of electricity. In order to achieve high thermal efficiency of the power generation cycle, the pressure and temperature of the steam must be maintained as high as possible. This, however, creates very difficult engineering problems which have not yet been successfully solved in the "conventional" reactors using water (light or heavy) or compressed gas as a coolant medium. The use of thermal reactors for generation of low temperature (and low pressure) process steam was considered not practical owing to the difficulty of ensuring steady consumption of this steam. In general practice the required quantities of process steam cannot justify the construction of large reactors. Atomic power stations are connected to a country-wide grid and take over the base load. A nuclear reactor for generation of process steam will not have generally such favorable conditions of operation and will be subject to fluctuations of steam consumption.

A nuclear reactor producing low temperature steam for distillation of sea-water for drinking and irrigation purposes, especially in arid and semi-arid countries, with the possibility of storing fresh water during reduced consumption, may have highly favorable conditions of operation. A "Low Temperature Reactor" (LTR), working in conjunction with an evaporation still, may have, under such conditions, a load factor equal to the availability factor which in this case may be taken as 95 per cent.

Steam for a flash type evaporation still usually has an inlet pressure between 10 and 15 psia and corresponding inlet temperature between 193°F and 213°F. The specified quantity of seawater demineralized per unit of low-pressure steam fed to

TABLE I

Comparative Data for the Shippingport Reactor and the LTR.

Item	Unit	Shipping-port Reactor (power)[1]	LTR (demineral)
Reactor heat output	B.t.u. per hour	790×10^6	790×10^6
Steam pressure	lb per sq. in abs.	600	15
Generator (heat exchanger) outlet steam temp. (t_2)	°F.	486.3	213
Steam generator feedwater temperature (t_1)	°F.	325	213
Temperature rise in steam generator (t_2-t_1)	°F.	161.3	0
Enthalpy rise (h_2-h_1)	B.t.u. per lb.	906.2	870.3
Steam generated ..	lb per hour	870×10^3	815×10^3
Overall fuel enrichment	per cent uranium 235	1.1	1.1
Coolant:			
(*a*) Substance ...	—	Light water	Light water
(*b*) Primary outlet temperature (t_4)	°F.	542	268.7
(*c*) Primary inlet temperature (t_3)	°F.	508	234.7
(*d*) Temperature rise in the core	°F.	34	34
(*e*) Temperature difference at heat exchanger coolant entrance (t_4-t_2)	°F.	55.7	55.7
Primary circuit operating pressure (pressure vessel)	lb per sq. in abs.	980	41
Design vessel pressure	lb per sq. in abs.	2,500	100

the evaporator varies considerably from one manufacturer to another. The figure of 6 lb. of fresh water per lb. of steam is considered safe. Safe future development may be expected to reach 8 to 9 lb. of water per lb. of steam.

FEATURES OF THE LOW TEMPERATURE REACTOR

The technological and constructional features of a pressurized-water reactor (PWR) will be modified when the plant is used for generation of low temperature steam. The important feature of PWR is that high-purity light water serves as both coolant and moderator operating under high pressure (2,000 psi in the Shippingport Reactor).

Temperature differences in the LTR will correspond to those of the Shippingport Reactor. The primary coolant temperature, which is 273°F lower than in the PWR, will greatly affect the pressure vessel—the design pressure of the LTR vessel will be 25 times lower than that of the Shippingport Reactor.

The Shippingport Reactor vessel has an overall height of 33 ft., with a welded cylindrical section of about 7 ft., and a nominal wall thickness of 8½ in. The hemispherical closure is forged and has final thickness of approximately 10 in. with an adequate reinforcement for the control rods and fuel tubes.[2]

The pressure-vessel design code requires consideration for stresses due to pressure, mechanical loading, and thermal gradients. A PWR vessel working in conjunction with a power plant is particularly susceptible to thermal stresses due to load variations.[3] The neutrons and gamma rays evolved produce in the walls of a reactor vessel an additional thermal gradient. The LTR, however, working in conjunction with a sea-water evaporation still, being steadily loaded and having thinner walls, will be subjected to lower thermal stresses since these decrease with wall thickness.

The LTR lower coolant temperature will favorably affect the corrosion rate of the vessel and piping, as the corrosion increases exponentially with temperature. To resist corrosion under the prevailing temperatures in the Shippingport Reactor, the vessel has a 0.25 in. stainless-steel inside cladding tightly bound to the carbon steel plates. The stainless steel protecting cladding may become superfluous in the construction of the LTR.[4, 5]

One of the main components of the PWR is the "pressurizer," which regulates the system pressure during load variations. Under any load variations within the prescribed limits, the phase of the coolant must not change. The LTR will enjoy a steady state of operation without any abrupt changes, thus no changes of temperature and pressure of the coolant may be anticipated. A similar pressurizer may be envisaged for the LTR, the major function of which will be reduced to that of a relief valve.

Many other primary system components will be either simplified or reduced in size. With considerably reduced danger of leakage, as a result of low pressure, the construction of pumps, especially those for main coolant, will be greatly simplified. The same may be said about the main stop valve, steam generator, piping system, and the hydraulic and steam apparatus. There will be no need for any steam separators.

To restrict the spread of radioactivity in the event of casualty, the nuclear part of the Shippingport plant is completely enclosed with steel containers. The design pressure of these containers is slightly over 50 psi, and their total gross volume is 600,000 cu. ft. It may be expected that for the LTR steel containers will become unnecessary. It is stated that 2,500 psia represents about the highest practical pressure with present-day technology. This may be compared with the 100 psi design pressure that is required for the LTR vessel.

DETAILS OF NUCLEAR BEHAVIOR (PRELIMINARY CALCULATIONS)

It is assumed that in steady operation of a PWR the fuel-element surface temperature will be the same as, or very close to, that of the coolant water. The heating-element surface temperature in the Shippingport Reactor is 88°F higher than the coolant exit temperature. It is suggested that the LTR core should have about the same size and geometry as in the Shippingport Reactor.

TABLE II—*Thermal Neutron Velocity*

Item	Unit	Shipping-port Reactor	LTR
Element surface temperature	Degrees Kelvin	605.2	453.4
Most probable velocity (v_0)	Cm per sec	3.159×10^5	2.725×10^5
Average velocity (v)	Cm per sec	3.552×10^5	3.066×10^5

The approximate evaluation of the neutron flux in the LTR is based on the following argument. When the neutrons are in thermal equilibrium with the moderator's atoms they will be subjected to a Maxwellian distribution of velocity. The most probable velocity v_0 corresponds to the kinetic energy kT, where k is the Boltzmann's constant ($=1.380\times10^{-16}$ erg per degree C) and T, the temperature of the distribution on the Kelvin scale.[6] Thus

$$\tfrac{1}{2}mv_0^2=k\mathrm{T} \quad \ldots\ldots (1)$$

$$v_0=\sqrt{\frac{2k\mathrm{T}}{\mathrm{m}}}\ \text{cm per sec}$$

$$=1.28\times10^4\sqrt{\mathrm{T}}\ \text{cm per sec.}$$

The average neutron velocity v of the Maxwell distribution is greater than the most probable velocity v_0 by the factor $2\sqrt{\pi}(=1.128)$, so

$$v=1.44\times10^4\sqrt{\mathrm{T}}\ \text{cm per sec} \quad \ldots\ldots (2)$$

The average thermal neutron flux (ϕ) in the Shippingport Reactor is about 5.9×10^{13} per sq. cm per sec.[2]

The thermal neutron density n at the average velocity can be evaluated from the equation

$$\phi=nv \quad \ldots\ldots (3)$$

TABLE III—*Thermal Neutron Flux (Tabulation)*

T (deg K)	$v_0\times10^5$ (cm per sec)	$v\times10^5$ (cm per sec)	$n\times10^{-8}$ (cm^{-3})	$\phi\times10^{-13}$ (cm^{-2} sec^{-1})	Reactor
605.2	3.159	3.552	1.661	5.9	Shipping-port
580	3.092	3.477	1.628	5.66	
560	3.038	3.417	1.603	5.48	
540	2.984	3.355	1.577	5.29	
520	2.928	3.293	1.551	5.12	
500.5	2.872	3.230	1.524	4.92	
475	2.798	3.147	1.489	4.69	
453.4	2.734	3.074	1.458	4.48	LTR

The thermal neutron density in the Shippingport reactor:

$$n=1.661\times10^8 \text{ per cu. cm.}$$

The thermal neutron density and the average neutron flux in the LTR can be (approximately) evaluated from the Maxwell distribution of velocity equation

$$dn=\frac{4n}{v_0^3\sqrt{\pi}}v^2e^{-\frac{v^2}{v_0^2}}\,dv \quad \ldots\ldots (4)$$

(a) Burn-Up

The "burn-up" is defined as the proportion of the total fuel in the reactor (including any fertile material that may be present) that is consumed in a given time and under given thermal neutron flux. Thus, we have [7]

$$\mathrm{B}=\mathrm{E}(1-e^{-\sigma\phi t}) \quad \ldots\ldots (5)$$

where B is the burn-up of uranium atoms (per cent), E is the uranium 235 enrichment (per cent), σ is the thermal neutron cross-section of uranium 235 (cm^{-2}) (assumed constant), ϕ is the thermal neutron flux (cm^{-2} sec^{-1}), and t is the irradiation time in seconds.

The depletion of the uranium 235 is considered to be influenced by fission only. The overall fuel enrichment in the Shippingport and in the LTR is 1.1 percent.

Let suffixes (1) and (2) relate to the Shippingport Reactor and the LTR, respectively. Letting $\mathrm{B}_1=\mathrm{B}_2$, $\mathrm{E}_1=\mathrm{E}_2$, and assuming σ=constant we get

$$\phi_1t_1=\phi_2t_2 \quad \ldots\ldots (6)$$

$$\frac{t_2}{t_1}=\frac{5.9}{4.48}=1.317 \text{ (refer to Table III).}$$

Under equal conditions, the heating element in the LTR will last about 32 percent longer than in the Shippingport Reactor. This prolonged life will have an important bearing on the fuel economy of the LTR.

(b) Power Output

The fission cross-sections of uranium 235 at low neutron energies range, vary in accordance with the "1/v law," that is, the fission cross-section will inversely decrease with thermal neutron velocity. Consequently, the fission cross-section in the LTR will be higher than in the Shippingport Reactor.[8]

The reactor power output P in kW may be expressed

$$\mathrm{P}=8.3\times10^7g\sigma_f\phi\mathrm{kW} \quad \ldots\ldots (7)$$

where g is the mass of fissile material in gm., σ_f is the fission cross-section, and ϕ is the neutron flux.

According to the 1/v law, we get

$$\frac{\sigma_{f1}}{\sigma_{f2}}=\frac{3.074}{3.552} \text{ (refer to Table III)}$$
$$\sigma_{f2}=1.15\sigma_{f1}$$

In order to get equal power output from both reactors we set $\mathrm{P}_1=\mathrm{P}_2$:

$$8.3\times10^7g_1\sigma_{f1}\phi_1=8.3\times10^7g_2\sigma_{f2}\phi_2$$
$$g_2=1.145g_1$$

thus, the fuel inventory of the LTR must be by 14.5 percent greater.

(c) Reactivity

With the decrease of temperature the reactivity of the core will increase. The increased reactivity will adversely affect the nuclear stability of the reactor, thus requiring more attention to the control

system. However, this consideration is of major importance only in connection with transient changes of the power level. The LTR working in conjunction with an evaporation still, as already mentioned above, will have a steady power level, practically without fluctuations.

(d) Xenon Poisoning

Xenon 135 is the most important fission-product poison. During the steady work of the reactor an equilibrium xenon 135 poisoning is created. An excess of reactivity is required in order to compensate the effect of this (as well as other) poisoning. The xenon poisoning increases with thermal neutron flux. In the LTR the equilibrium xenon poisoning will be lower than in the Shippingport Reactor.

(e) Conclusion

Certain nuclear features of the LTR were shown to compare favorably with those of the Shippingport Reactor.

ECONOMY OF SEA-WATER DISTILLATION

There is little experience in the construction and operation of large evaporation stills. The largest sea-water demineralization plant, using exhaust steam from a 12,500 kW turbine, is now being erected in Aruba, Antile Islands. The plant will supply about 10,000 cu. meters per day of distilled water, using six-effect evaporators. Although multiple-effect evaporators for sea-water demineralization are extensively used on board ships, their size and heat economy are in no way suitable for large-scale water supply. Recent research in problems of drop-wise condensation, thin turbulent liquid films, and the optimum velocities of the interacting agents (steam and sea-water) as well as the development in multi-stage flash-type evaporators will undoubtedly considerably increase the distillate to steam ratio in the evaporating stills.

The Saline Water Conversion Report, 1955, is quoted: "Research investigations to date have demonstrated that heat transfer rates as high as five to seven times that attained in present-day evaporators are readily maintained." [9]

In distillation of sea-water an important item of expense is the cost of steam, that is, the cost of fuel per unit of demineralized water. Since for distillation low-pressure steam (around 15 psia) is used, the process may be economically performed using exhaust steam from steam turbines working against back pressure instead of vacuum otherwise created in the condenser.

A question may be put forward: under what conditions will it be advisable to use a LTR, generating low-pressure steam for the single purpose of distillation of sea-water, as compared with a dual-purpose PWR (Shippingport type) generating electricity and using exhaust steam for distillation?

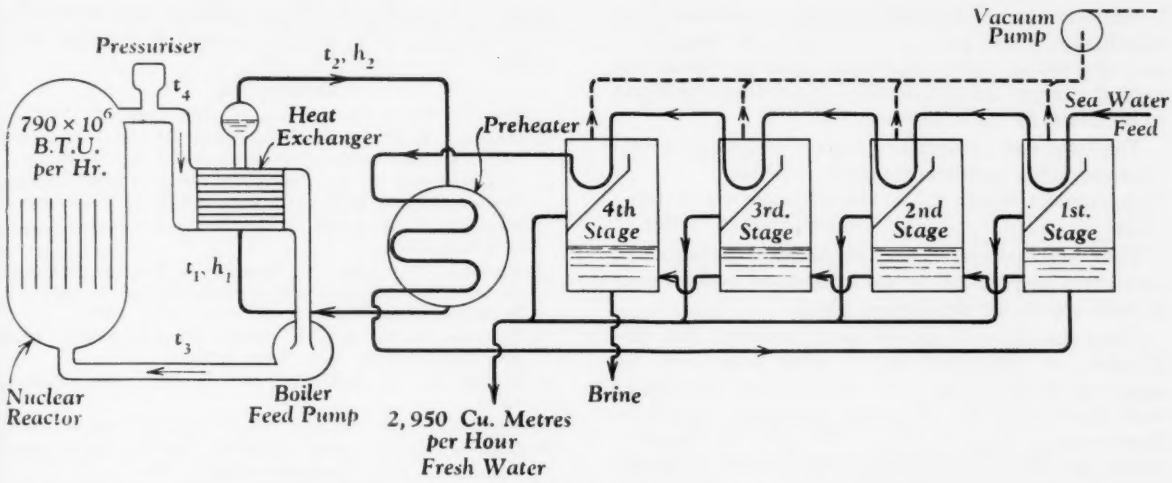
It can be shown that the generating capacity of the Shippingport Reactor working on a dual purpose 15 psia extraction turbine will be reduced from 71,400 kW to 55,000 kW. The quantity of steam which will be directed in this case to the evaporator will be 600×10^3 lb. per hour. With safe rating of the evaporation still, the Shippingport Reactor will distill 2,170 cu. meters per hour as against the 2,950 cu. meters per hour of the LTR. Taking into account 75 percent load factor of the power plant, the yearly production of distillate will be 14.25×10^6 cu. meters, as compared with 24.55×10^6 cu. meters from the LTR. It is assumed that the LTR evaporation still will operate 8,300 hours per year (95 percent load factor).

TABLE IV—*Distillate to Steam Ratio*

Producer	Daily supply, cu. metres per day	Distillate to steam ratio	—
Continental firm	50,000	4.65	Project, using steam from a 100,000 kW power station. (Private communication)
G. and J. Weir	8,000 to 10,000	4.88	Aruba, Antilles Islands, using exhaust steam from 12,500 kW power station
The Griscom Russel Co., U.S.A.	18,500	6.0	Project J. Leicester, *British Chemical Engineering*, Aug., 1956
W. L. Badger	65,000	20.0	Project Saline Water Conversion, 1955, Annual Report
Safe future development	50,000 to 100,000	8	

The single-purpose LTR will be suitable for localities where the requirements for sweet water are larger than the amount which can be supplied by using exhaust steam from power plants, that is, in arid regions situated on the sea shore or in its proximity, where the requirements for electric energy or the cost of fuel do not warrant the erection of suitable power plants. This applies especially to industrially underdeveloped countries. The following may serve as an example: the irrigation of 1 acre in an arid zone requires 3,000 cu. meters of water per annum. An efficient evaporation still will require the exhaust steam of a 1 kW turbine to produce about 380 cu. meters of distilled water per annum working with 95 percent load factor. For the irrigation with distilled sea-water of a tract of land of say 75,000 acres, exhaust steam from a 600,000 kW capacity power plant will be required. This tract of land is sufficient for 8,000 to 9,000 smallholding farms with an agricultural population of only 40,000 to 50,000. A power plant of such capacity will certainly be out of the question.

The following basic data were taken for the comparative cost evaluation of a dual-purpose atomic plant (generation of electricity and distillation of sea-water) and a single-purpose atomic plant (distillation of sea-water only). There are no data available for the Shippingport pressurized-water reactor. We shall use the average figures given last year by Sir Christopher Hinton, relating to an



Low temperature reactor (LTR) for sea-water demineralization. A reactor of this kind using light water as coolant and moderator will operate at a pressure which is a fraction of that in a conventional power-producing pressurized-water reactor (PWR).

atomic power plant of 150 MW capacity.[10] It should be noted that the load factor of atomic power plants is likely to decrease as their share in carrying the grid load increases.

TABLE V—*Cost of Electricity Generated by an Atomic Power Plant*

1(*a*) Capital charges, nuclear portion	2.85*	
1(*b*) Capital charges, electric generation portion	1.50	0.00435 dol per kWh
2. Fuel inventory		0.00070 dol per kWh
3. Fuel replacement		0.00280 dol per kWh
4. Operation and maintenance		0.00070 dol per kWh
		0.00855 dol per kWh

* Division of capital charges taken from other sources.
Load factor 75 percent, grid price 0.0070 dol per kWh. No residue value of the fuel element. No credit for plutonium. Investment in the plant (nuclear and electric generation portions), 350 dol per kW.

The investment in the nuclear portion (including steam generation) of the LTR per kW (heat) will be considerably lower than that in a PWR. For a reactor of 500 MW (heat) capacity the investment of 30 dol per kW (heat) is considered sufficient. Although the LTR dealt with in this paper is of a smaller size—about 225 MW (heat)—we are using, for sake of comparison, the above figure of 30 dol per kW (heat). The conclusions arrived at will be also adaptable to larger reactors.[11]

As has been shown, the LTR will require a 14.5 percent increase of fuel inventory, and the life-time of its fuel elements will increase by 32 percent as compared with the Shippingport Reactor. The operation and maintenance of the LTR (distillation only) will be also reduced.

The overall thermal efficiency of the Shippingport power cycle is 30.1 percent. Recalculating Table V per kWh (heat) generated in the steam generator, and making the necessary adjustments for the LTR working with a 95 percent load factor, we arrive at the figures in Table VI.

TABLE VI—*Comparing Cost of Steam Heat Generated, mills per kWh (heat)*

—	PWR 75 per cent l.f.	LTR 75 per cent load factor	LTR 95 per cent load factor
1. Capital charges (nuclear portion)	0.858	0.37	0.29
2. Fuel inventory	0.210	0.24	0.19
3. Fuel replacement	0.840	0.57	0.45
4. Operation and maintenance	0.210	0.12	0.10
Total	2.118	1.30	1.03

The decreased income from the Shippingport Reactor working on dual purpose may be calculated as follows:

Decreased generation per year: $(71{,}400-55{,}000) \times 8{,}760 \times 0.75 = 108 \times 10^6$ kWh.

Decreased income per year: $108 \times 10^6 \times 0.00855 = 928{,}000$ dol.

This decreased income is a direct result of the reactor demineralizing 14.25×10^6 cu. meters of sea-water per annum. The cost (not including the distillation portion) per cu. meter of distillate will be: $(928{,}000 \div 14.25) \times 10^6 = 65.2$ mills per cu. meter.

Heat required in the LTR per lb of steam (Table I) will be: $\frac{790 \times 10^6}{815 \times 10^3} = 970$ B.t.u. per lb which is equal to 78.3 kWh (heat) per cu. meter of distillate.

According to Table VI, the LTR steam heat cost per cu. meter of distillate will be: $1.03 \times 78.3 = 80.6$ mills.

Various sources indicate that the investment in a distillation plant per cu. meter-year, is roughly 1 dol, depending upon size and capacity. Thus, the capital charges for the distillation portion are found to be about 80 mills per cu. meter.

The total cost of distilling water, including capital charges, labor and maintenance, will be:
Shippingport Reactor ....175 mills per cu meter
LTR .....170 mills per cu. meter.

The only practical and economical solution of sea-water conversion is an integrated power generation-sea-water distillation plant.

From the above it can be seen, however, that distillation of sea-water on a large scale can be achieved at a similar cost (as a matter of fact at even lower cost) by using heat generated by a Low Temperature Reactor. This will make the application of the distillation process independent of power development.

The cost of sea-water conversions is heavily influenced by the cost of the distillation equipment. The development of big, industrially sound, distillation plants is in its initial stages. It may thus be expected that, in due years, cost of equipment will trend downward, contributing along with more efficient processes, to a lower price of distilled sea-water.

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Working with George G. Sharp, Inc., the General Electric Company has recently completed a preliminary design study of a 22,000 SHP nuclear propulsion plant for a T-5 type tanker. The design is based on the use of a boiling water reactor. By accepting certain disadvantages, the principal of which is a limited access engine room, several improvements in PWR characteristics appear to be possible. One of the principal improvements is in the economic area. The study indicates a considerably lower first cost for the machinery plant than that of the SAVANNAH type installation, and a fuel cost closely approaching that of fossil fuels. Another improvement appears to be lighter weight—the BWR scales out at about 185 pounds per horsepower. Dry, saturated steam at 1000 psi is contemplated. Two papers describing the study was presented at the Second Annual Nuclear Merchant Ship Symposium, sponsored by the Maritime Administration. The papers also dealt with problems of control, shielding, waste disposal, and ship damage hazards.

—from MARINE ENGINEERING/LOG

October, 1958

HOLMES F. CROUCH

REFUELING NUCLEAR SHIPS

THE AUTHOR

is a graduate of the U. S. Coast Guard Academy in 1940, served on various Coast Guard vessels during World War II, and later (for seven years) as a safety inspector of merchant ships. Subsequent thereto he has specialized in engineering research and technical writing in the electronic, nuclear, and guided missile fields. He holds a Master of Science Degree in Nuclear Engineering from the University of California. Mr. Crouch is presently employed as a nuclear engineer in the Research Division of the Marquardt Aircraft Co., Van Nuys, California.

REFUELING a nuclear ship is likely to be the most hazardous operation of its routine service. This is because, after many nuclear voyages, radioactive fission products accumulate within the reactor fuel elements. When removing these "spent" fuel elements, considerable care must be exercised to avoid releasing the fission residues and to keep workmen clear of radioactive areas. As a consequence, the amount of care required is significant in time, equipment, and manpower. The resulting cost is appreciable.

The combination of high cost and high hazard potential in nuclear refueling spells major departures from oil ship refueling concepts of the past. Instead of conventional refueling in a few hours using ordinary equipment such as oil hose and pumps, attended by a few members of the engineroom gang, nuclear refueling will impose a complexity of new procedures, new equipment, and new talents. Because of the greater in-port time required, the frequency of nuclear refuelings may be compromised to coincide with other major overhaul and repairs, such as drydocking, annual inspections, etc.

Needless to say, the salient features of nuclear ship refueling have not yet come to the fore and standard practices are undeveloped. In the meantime, it is helpful to review the basic principles—and hazards—involved when removing the spent fuel and replacing it with new fuel. An understanding of these principles could aid in the solution of the technical, economic, and social problems ahead.

Lethality of Spent Fuel

Spent fuel is a lethal substance. Every gram of fission residue contained in the spent fuel produces—on the average—about 500,000 curies of gamma radioactivity (see Figure 1). Keeping in mind that 600 roentgen of radioactivity over the entire body would kill a man, the conversion from curies to lethal dose rate is obtained from the following approximate relationship:

$$\text{lethal dose rate} \approx \frac{CE10^{-2}}{d^2} \text{ r/hr}^*$$

where, C = radioactive strength in curies; E = energy of radioactive disintegration in Mev; d = distance from radioactive source in feet.

*S. Glasstone, *Principles of Nuclear Reactor Engineering*, p. 545, p. 555.

Thus, one *gram* of spent fuel residue is capable of killing a man (when exposed for one hour) at a distance of fifty feet.* This presumes that the radioactive species are clumped together in a bare point source, without a shield of any kind.

To appreciate the magnitude of lethality for an operational nuclear ship, suppose the refueling cycle is every 18 months. Suppose further that we have a 60 megawatt ship reactor (about the size of a large submarine or a Mariner-type merchant ship reactor) which is underway 80 percent of the time. Every megawatt-day of steaming consumes approximately one gram of fissionable fuel, and in so doing produces a like amount of radioactive residue. In the course of one refueling, our 60 MW ship would have produced 30,000 grams (65 pounds) of fission residue. This would be distributed throughout all of the fuel elements which, because of their cladding, would self-absorb, perhaps, half of the gamma radiation emitted.** Even so—if the spent fuel elements were to be unloaded en masse from the reactor—the lethality of escaping radiation could kill hundreds of men, if unprotected. It is emphasized that this is strictly a possibility which rarely, if ever, would happen. One nuclear ship already has refueled without incident, and others will, too, in the future.†

Importance of Aftercooling

When a nuclear ship reactor is shut down in preparation for refueling, the fission product residues generate an appreciable quantity of heat on their own.

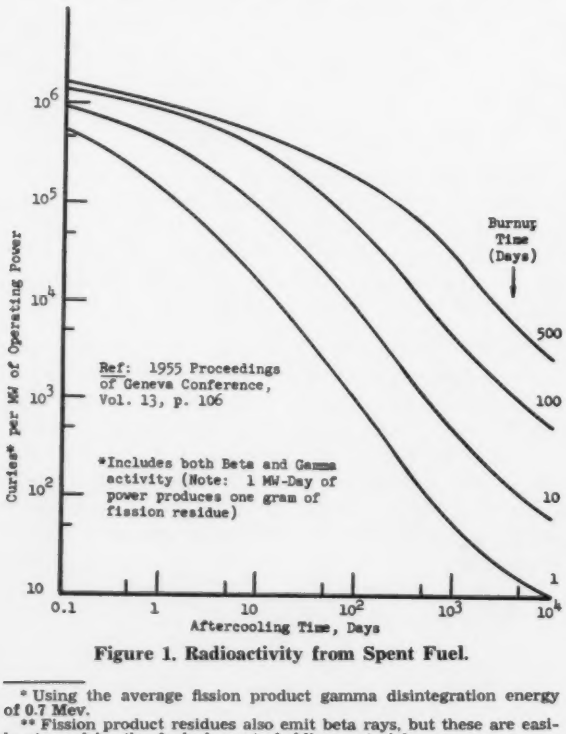
Curies* per MW of Operating Power
Burnup Time (Days)
500
100
10
1
Ref: 1955 Proceedings of Geneva Conference, Vol. 13, p. 106
*Includes both Beta and Gamma activity (Note: 1 MW-Day of power produces one gram of fission residue)
Aftercooling Time, Days

Figure 1. Radioactivity from Spent Fuel.

For our 60 MW ship again, operating 18 months, this "decay heat" (as it is called) approximates 2.5 MW . . . or about 9 million btu/hr.* This decay heat continues to be generated, at a slowly diminishing rate, for many days after shutdown. For example, three months after shutdown, the total heat generation would approximate 200,000 Btu/hr—still an appreciable amount (see Figure 2).

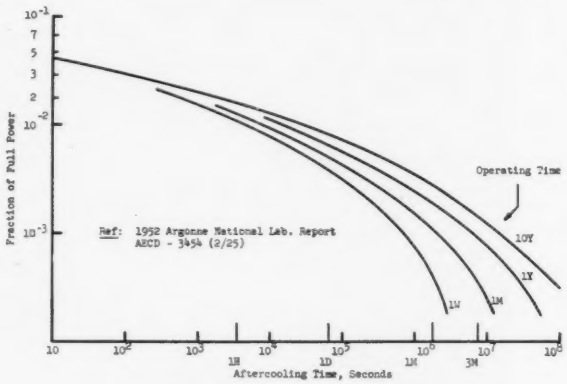

Figure 2. Decay Heat from Spent Fuel.

Now, the importance of decay heat is this: if the heat were not removed continuously, the fuel elements would melt down. This would release the radioactive fission residues to the surroundings. Many of these residues are gaseous; others are microscopic particulates. If, for example, meltdown were to occur in the reactor compartment, the gaseous residue would disperse towards the ship's structure overhead (framing, piping, wiring, rigging, etc.) and the particulates would disperse towards the decking, bulkheads, hatchways, gangways, etc. The result would be a widespread area of radioactive contamination.

To prevent the meltdown of spent fuel elements, aftercooling is required. This must be an auxiliary cooling system (water is the usual cooling medium used) independent of the main reactor cooling system. The problem is that the aftercooling system must be an integral part of the spent fuel element package from the time of reactor shutdown until the fuel elements have been sufficiently cooled to make meltdown impossible. The aftercooling time required would depend on the fuel element design, on the reactor power, and on the ship's operating history. For some reactor types, aftercooling may be required *while* the spent fuel elements are being removed from the reactor, *while* they are being transferred from the ship's reactor compartment to dockside, and *while* they are being shipped from dockside to other transfer areas or to remote reprocess plants.

* Using the average fission product gamma disintegration energy of 0.7 Mev.

** Fission product residues also emit beta rays, but these are easily stopped by the fuel element cladding material.

† The Nautilus was refueled after steaming 62,000 nautical miles in a period of 26 months (see p. 18 of *Nucleonics*, May, 1957).

* MW = 3.413 million Btu/hr.

Preparation for Defueling

Preparation for defueling requires a lot of planning and organization. Defueling personnel must be shift-organized so that in the event of a mishap a new crew can substitute quickly so as to time-limit the radiation exposure of all. Monitoring personnel must be on hand with detection instruments and the instruments must be in ample supply so that, in the event of instrument contamination from leaks and spills, replacements can be promptly made. Reactor dismantling equipment must be in good order with standby spares and emergency power, in the event of equipment contamination and utility failures. Spent fuel handling equipment must be rugged, fail-safe, and precision aligned so that once the defueling operation begins, it can be carried to completion in minimum time . . . with minimum risk to personnel.

Rather than attempting to describe defueling preparations in detail, a partial checklist of defueling functions and equipment is given in Table 1. Necessarily, these functions would be expanded considerably and tailored to each particular nuclear ship and its particular reactor design. As the fleet of nuclear ships grows and defueling experiences unfold, standardized procedures undoubtedly would evolve.

Because of the mandate for safety and the specialized personnel training required, defueling functions would rarely be performed by a ship's engineroom crew. The preparational planning—as well as the actual work of defueling—would be done by shoreside personnel. Even if shipboard personnel had the experience, one ship's "black gang" would not be enough. It requires from three to five times the number of workers and amount of equipment to perform a nuclear defueling operation as it does to perform a similar operation if no radiation were present.* The paragraphs which follows may indicate some of the reasons why.

Five Dismantling Procedures

Present reactors for ships are of the pressurized water type—called PWR. A representative sketch of such a reactor is shown in Figure 3.** Let's typify a PWR reactor and say that the water (which serves both as a reactor coolant and neutron moderator) is under 2000 psi pressure. Let's describe the reactor further as being 33 ft. high (shell only), 9 ft. diameter, with a shell thickness of 9 inches. This shell houses the reactor core in the form of a right cylinder, 6 ft. diameter by 6 ft. in height. This core consists of, say, 25 bundles of 100 fuel elements each (2500 fuel elements total). Interspersed among these 25 bundles are, say, 15 control rods. The entire reactor is surrounded by a 3 ft. water shield, with 9 inches of lead. The problem, now, is to go in and remove the spent core.

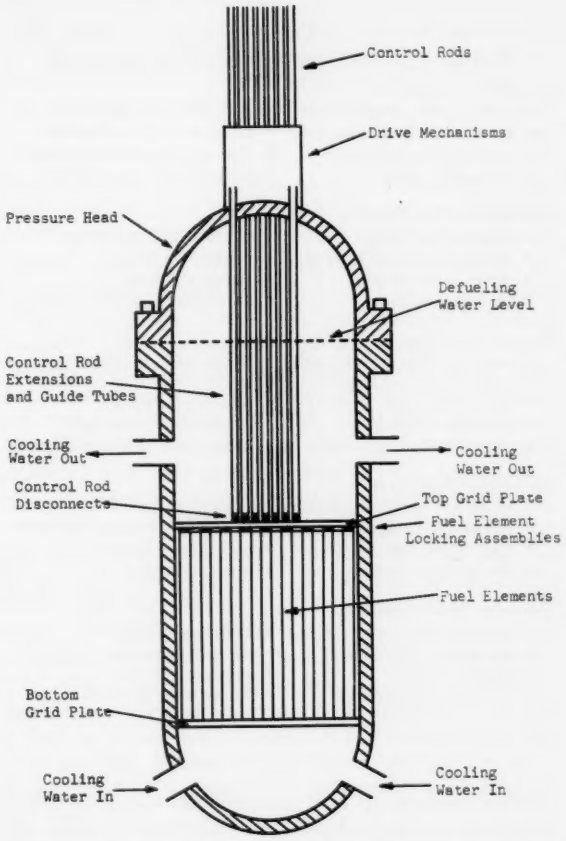

Figure 3. Typical PWR Showing Fuel Elements and Control Rods.

Assuming that the necessary personnel and equipment are available, the first procedure would be to have the dismantling and defueling personnel don *two* suits of overalls, then leggings, overshoes, full face masks, and two pairs of gloves. Personnel who are not going to take part in the operation should be cleared out. Radiation exposure time limits should be established, with prearranged (audible) timing signals for reliefs.

Procedure two (assuming aftercooling operation) would be to disconnect the control rod drive mechanisms. Clearing away the 15 drive mechanisms—with their power connections, seals, covers, scram trips, instrument wiring, etc.—is a significant chore in itself. This actuation equipment must be completely removed so that the rods are free inside of their guide tubes, inside the reactor. The disconnected mechanisms are radiation monitored, then set aside for repair or replacement.

Procedure three is to remove the shielding top plug, shielding access, or other covering from the control rod housing and reactor head. This involves hand-hooking or remote-hooking onto overhead

*K. K. Campbell, "Maintenance Work in the Field of Nuclear Energy," 1953 Conference on Nuclear Engineering—University of California, Berkeley.

**Simpson and Shaw, "PWR Power Plant," *Westinghouse Engineer*, November, 1955.

TABLE 1
Partial Checklist for Nuclear Ship Defueling

Defueling Personnel
- adequately trained
- standbys available
- protective clothing
- exposure history
- specified assignments
- badges and dosimeters
- changeroom procedures
- emergency procedures

Monitoring Personnel
- monitoring plan
- instruments calibrated
- time limits posted
- warning signs
- control access areas
- roped-off areas
- waste disposal
- emergency procedures

Dismantling Equipment
- remote disconnect tools
- scavenging equipment
- drip pans and buckets
- portable shields
- aftercooling facilities
- lights and microphones
- periscopes and mirrors
- top-plug storage
- control rod storage
- tongs and manipulators

Defueling Equipment
- shielded defueling crane
- defueling coffins
- extraction tools
- underwater lights
- closed circuit TV
- underwater manipulators
- emergency power
- area monitors

Transfer and Cleanup
- cooling pits
- transport facilities
- danger signs
- cleanup equipment
- absorbing paper
- decontamination of tools
- reactor repairs
- reloading approval

cranes, freeing up bolt connections and gear teeth, breaking control rod pressure and thermal seals, and storing the shielding and covering in a nearby roped-off area. The top shield would have a certain amount of induced radioactivity in it, though this would not be a major hazard.

Procedure four—and this becomes the first really difficult step—is to remove the 9-inch thick pressure head or one of the pressure head access ports, depending on the reactor design. Anywhere from 10 to 40 six-inch studs would have to be freed . . . but not by hand! Remember, the top shield has been removed. The stored-up pressure stresses (2000 psi) in these studs would have to be relieved by electrical or pneumatic impact wrenches, controlled and operated from behind portable shields by workers looking through periscopes, corner mirrors, and listening to speakers from microphones attached to the wrench heads. Locating and freeing bolt studs in this manner is no ordinary task, to say nothing of remote hooking-on to pressure head hoisting lugs when the studs are clear. Removing the pressure head (or access port) exposes the control rod guide tubes, and leaves the spent core staring up through several feet of water with an eerie-blue radiation (Cerenkov) glow.*

*When radiation traverses a transparent medium (e.g., water or air), it emits a "Cerenkov glow" when its velocity is greater than the speed of light. See Gordon and Hoover, "Burst-Slug Cerenkov Detector," *Nucleonics*, January 1957, pp. 91-92.

At this point, procedure five, underwater lights are lowered into position to focus on the core grid plates, on the fuel element locking assemblies, and on the control rod guide shafts. A closed circuit TV camera or other remote viewing equipment is rigged in place. An overhead crane—with the operator in a heavily shielded cab, looking through shielding windows and at a TV screen—lowers remote manipulator tools and tongs into the brilliantly clear, core water.* The manipulators are actuated by workmen on the deck (behind shields) who disconnect the control rod guide shafts, leaving the poison portion of the control rods to safeguard the spent core. The guide shafts are removed and stored in a special shielded container. Then a fuel element extraction tool is lowered into place . . . in preparation for the crucial act.

Precision Withdrawal a "Must"

The extraction of spent fuel elements must be done carefully—purposely—with the skill, precision, and determination of a dental surgeon. The extraction tool is latched on to a specifically designated bundle or bank (several bundles) of fuel elements. Lifting (withdrawal) force is applied . . . gentle at first, then a steady pull. At this point, it's too late to make adjustments or other preparations. The fuel elements must come out!

Once the force of extraction starts, the direction of motion must be up. No sidewise motion; no twisting—just straight up. There is good reason for this "one-direction-only" mandate. Some of the 18-month old spent fuel elements may be warped or slightly damaged; some may have small leaks; the cladding walls may be corroded thin. They could be so easily punctured by careless handling, with the result that fission product gases and particulates could readily escape. Hence, withdraw with precision.

Generally, some form of hydraulic extraction rig is used. The extraction tool is on an extendable shaft which reaches into the reactor and then recedes into a cylinder mounted on top of a 9-inch lead lined steel coffin. This coffin is precision indexed over the opened reactor head, or over the fuel access port (as the case may be). Often, the coffin, extraction shaft, and aftercooling apparatus form an integral piece of equipment (see Figure 4). When the fuel elements are inside the coffin, a sliding lead plug closes to provide total shielding.

The precision movements do not end here. The overhead crane which transfers the spent fuel coffin to dockside must lift vertically only, until the coffin is free and clear. Then it may be moved horizontally and laterally . . . with great care. The coffin, it should be pointed out, weighs on the order of 25 tons. One can't afford to bang this weight—and this hazard—around.

* Manipulator tools are mechanical arms and grippers with all degrees of human motion which can be remotely operated.

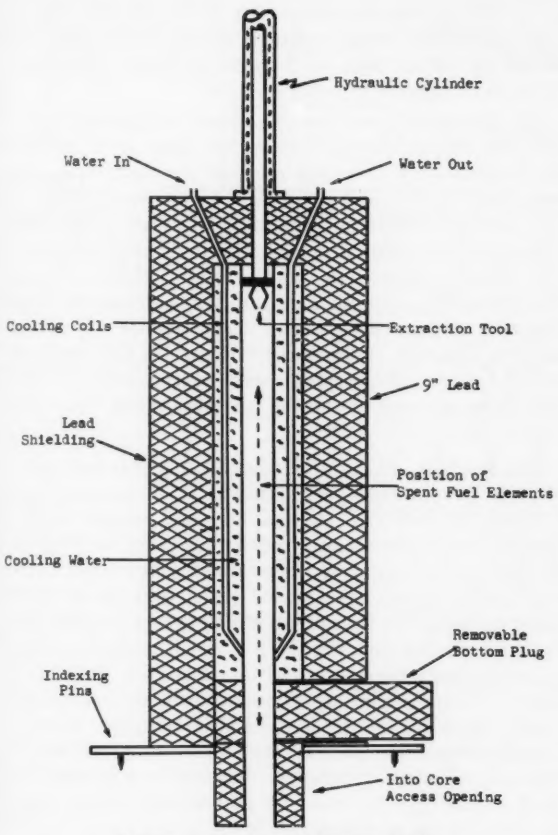

Figure 4. Typical Defueling Coffin.

Transfer to Reprocess Plants

At dockside, the fuel elements are unloaded from the coffins and reloaded into transfer pits. These transfer pits (synonymously called casks, cells, pools, etc.) are larger in size than the defueling coffins, less complicated, and less precisely designed. They are generally of steel and concrete construction with appropriate brackets and mountings for securing the fuel elements in the upright position. These pits are flooded with water which is circulated through an external cooling source. The size of these transfer pits could be made to accommodate a single bank of fuel elements . . . or an entire reactor core. Design care must be exercised, however, not to permit the storage of spent fuel elements in those arrangements which approach critical nuclear proportions. Otherwise, a live reactor would be on one's hands instead of a safely arranged stacking of spent fuel elements. Cadmium safety plates or safety rods, which gobble up stray neutrons, are often used to prevent nuclear criticality proportions.

The transfer pits may be mounted on railroad cars or on barges at the nuclear ship's side. These transport facilities may be routed to a remote off-site location to await further decay-cooling of the spent fuel elements. Then the long trek to the reprocess plants inland.

The trek to reprocess plants is a major undertaking in itself. Remember, we are dealing with a heavy lift package—on the order of 50 to 100 tons—and one that is potentially hazardous. The familiar "Do Not Hump" signs on railroad cars now take on a new significance. The spent fuel elements are in their same fragile condition as they were when they came out of the reactor. A wreck or careless handling of railroad cars could open up hairline cracks in the fuel element walls, thus releasing radioactive fission residues.

This possibility means that radiation monitoring equipment and personnel must accompany the fuel element transfer to the reprocess plants. These personnel must be alert to detect, report, and remedy any radiation leakage. Also, an armed guard should accompany the operation, in the event of a railroad accident. The role of the guards would be to keep inexperienced workers and curious people away.

At the present time, there exists only one nuclear ship spent fuel reprocess facility. This is the Navy's "Expended Core Facility" at the National Reactor Testing Station, Idaho Falls, Idaho.* This is the facility to which the 62,000-mile spent core from the *Nautilus* was shipped.** One of the principal functions of this facility is to cut open the spent fuel elements and examine them with the purpose in mind of finding better ways to advance the technology of nuclear fuels. Subsequently, the spent fuel is sent to nearby NRTS plants for reclaiming the *un*spent fissionable fuel.

Reloading New Fuel

Prior to reloading new fuel into a ship's reactor, considerable decontamination of equipment and area cleanup is required. This is because the defueling operation unavoidably contaminates that equipment used in contact with the reactor vessel and the spent fuel elements. The "contamination" is in the form of radioactive rust and radioactive water droplets which can be spread around with amazing ease. These radiocontaminants are not a lethal hazard, yet they do necessitate extensive radiation monitoring precautions. As a result, practically all of the dismantling and defueling equipment must be decontaminated before it can be used again. The same is true of the defueling personnel's clothing. They must take off their defueling clothing and put on clean clothing before starting the reloading operation.

The new fuel elements may be one of two types. They may contain all-clean (brand new) fuel material, or they may consist of reprocessed fuel material. Both types have equally good power generation capabilities. The difference is that the all-clean fuel elements could actually be handled by hand

*See p. 9 of pamphlet by the Idaho Falls AEC Office: "Thumbnail Sketch—National Reactor Testing Station."

**See pp. xiv and 60 of "Radiation Safety and Major Activities in the Atomic Energy Program," July-December, 1956, USAEC.

without shielding, coffins, or remote devices. The reprocessed fuel elements contain some—not much—fission product residue from the predecessor spent fuel. Thus, some shielding and handling precautions would be required. With either type of new fuel, however, the reloading operation is considerably less hazardous than the unloading, though the time consumed may be nearly as great.

Generally speaking, all fuel elements for a given reactor are the same size and shape, and therefore look the same physically. But their nuclear properties can be vastly different. In order to achieve as nearly uniform power generation in the reactor as possible (this increases total power, reduces thermal stress, minimizes hot spots, etc.), the fissionable fuel nuclei are dispersed in varying amounts. The fissile material in the center fuel elements of the reactor, for example, is "thinned out" so that the peak neutron and thermal fluxes do not burn up this section of fuel elements before all others. In other fuel elements, burnable poisons are added.* Consequently, the in-core arrangement of new fuel elements must be undertaken with planning and care. A typical reloading arrangement is shown in Figure 5.

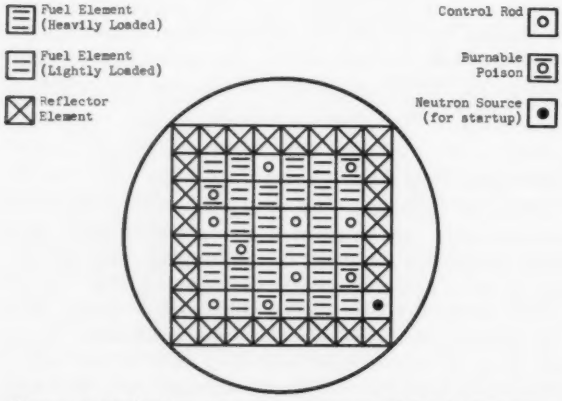

Figure 5. Typical In-Core Arrangement of Fuel Elements.

Future Refueling Depots

The length of time for, and cost of, a nuclear ship refueling is not of great concern at the present time. Only a few ships are involved. So, in the interests of safety and the development of a new ship propulsion technology, refueling is a secondary matter. It has been estimated that with a well-trained refueling organization the complete job could be done (perhaps) in a week's time. The cost, excluding that of the fuel, could approach (this is a guess!) a quarter-million dollars per refueling.

In future time of war, and even under peacetime competitive conditions, one wonders if there wouldn't be some way by which nuclear fleet refueling couldn't be speeded up, and cost-reduced . . . without sacrifice in safety. Yes, possibly, there are ways. But these require further evolution in refueling technology and accompanying modifications in shipboard reactor designs.

As discussed previously, present shipboard reactors are top unloaded. This, as we have seen, indeed is a satisfactory and safe method for defueling. An even safer way—and much less time consuming—is the method of *bottom* unloading. In other words, the spent fuel elements are pushed out through the bottom of the reactor through a water chute, where they drop into a water canal under the reactor. This practice is used very successfully at the Materials Testing Reactor, Idaho Falls.* Possibly, this method could be adapted to shipboard reactors. If so, the defueling procedure might go something like the following.

The nuclear ship would moor into a graving dock, and the water would be pumped out, in normal drydocking procedures. Adequately below the dock, there'd be a spent fuel transfer canal (see Figure 6). From this canal, a hydraulically operated caisson (with appropriate water locks and guide surfaces to keep the fuel elements upright) would rise up through the graving dock floor. Simultaneously, a hydraulically operated defueling port in the ship's bottom would open up. The graving dock caisson would then attach to a defueling adaptor permanently designed as part of the mounting structure of the reactor shell. Then the bottom of the reactor shell would open up making a complete U-tube water head between the water level in the reactor and that in cooling pits alongside of the graving dock (see Figure 6 again). It would then be a matter of remotely unlocking the spent fuel elements (by integral mechanisms in the reactor shell design) and letting them sink slowly to the canal floor. From here, they would be underwater transferred to designated cooling storage spaces where they would remain until all chance of meltdown was gone.

Concurrent with the defueling operation above, regular ship's bottom cleaning and painting, overhaul of fittings and discharges, and tail shaft and propeller inspections could be carried on. Concurrently, also, new fuel elements could be reloaded through the reactor's top. Thus, the complete cycle of nuclear refueling—including normal drydocking work—could be performed in a day or two—or three at most.

There's future possibility that such a refueling depot as just described could come into being. Because of the capital cost involved and the necessity for national security, it would be Government owned and, most likely, Government operated. Possibly, one such depot would be centrally located on both the East and the West Coasts of continental U.S. It would seem that for reasons of mutual tech-

*See "ETR: Core and Facilities," *Nucleonics*, March, 1957, pp. 44-47.

*Goertz and Bevilacqua, "Remote Handling—Reactor Refueling Systems," *The Reactor Handbook*, Vol. 2, USAEC-3646, pp. 855-859.

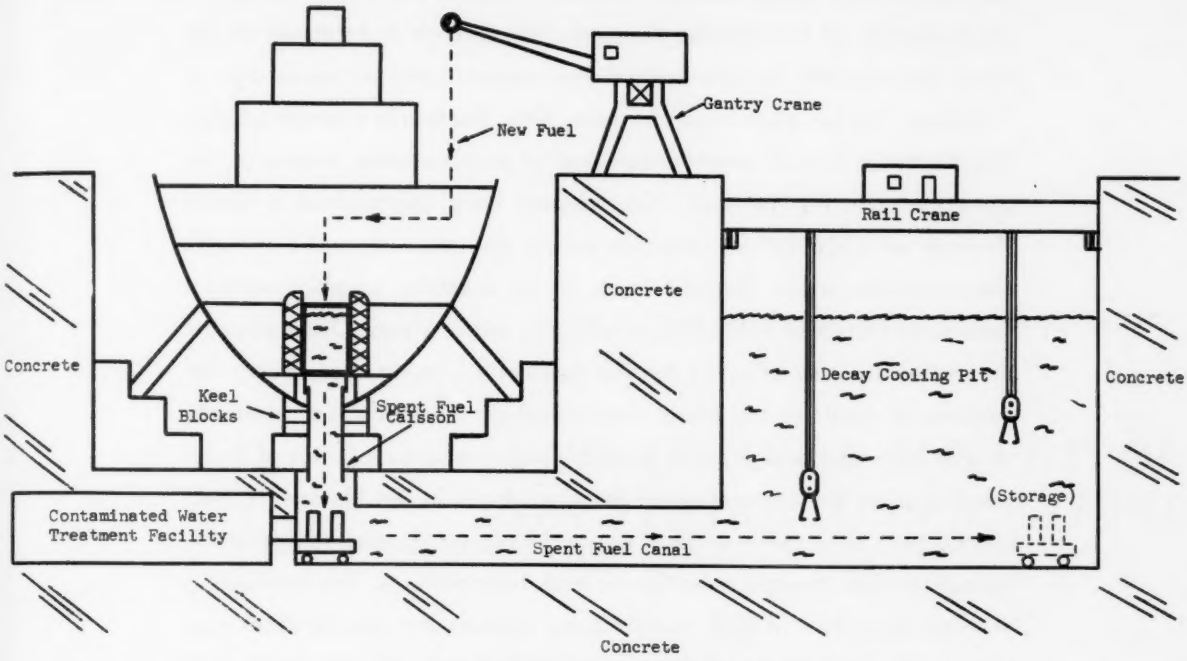

Figure 6. Possible Nuclear Ship Refueling Depot.

nical interest, sharing of costs, and nuclear safety, each refueling depot would serve both Navy and Merchant nuclear ships.

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Considerable attention has been focused in recent years upon the physical properties of fine metallic filaments. Mr. George A. Hoffman of The Rand Corporation has summarized the present state of knowledge of "whiskers." As has been known for some time, the tensile strength of very fine filaments is much greater than that of polycrystalline masses of the same material. For instance, iron whiskers have approached a tensile strength of 2,000,000 psi. Metallic oxides also show dramatic strength characteristics in the filament form. As an example, sapphire whiskers have been tested to 1,700,000 psi ultimate tensile stress. Other physical properties are also affected by this geometry, among them being the modulus of elasticity and creep characteristics. The modulus of elasticity may be increased or decreased from its normal value by a factor of 2, depending upon the orientation of the crystal axis in the filament. Creep has not been observed in whiskers below the recrystallization temperature, indicating high strength retention at high temperatures. The fabricating of large quantities of high performance whiskers into usable stock material is an intriguing possibility because of the great gains in physical properties which appear to be possible. Mr. Hoffman discussed various fabrication techniques and the future potential of this field.

—from JOURNAL OF METALS
September, 1958

The world's first commercial solar cell panel is being manufactured by Semiconductor Division of Hoffman Electronics Corporation, Evanston, Illinois. The panel is an assembly of 144 individual silicon cells, and is designed to supply five watts of electrical power. The sensitive cells are thin wafers of silicon with controlled amounts of arsenic and boron incorporated to improve the power conversion performance. Absorption of light energy by the discs causes an electron displacement, creating a voltage buildup in each cell. The specific application requirements are met by joining the cells in various combinations of sizes and shapes. The cells are not affected appreciably by variations in humidity and temperature, and do not deteriorate, having an estimated lifetime of about 10,000 hours. Their top efficiency of about ten per cent does not place the solar converters generally in competition with more conventional power sources. However, the converters fill a need for supplying low power requirements.

—from JOURNAL OF THE FRANKLIN INSTITUTE
October, 1958

"THE CHARTERED MECHANICAL ENGINEER"

VIBRATION: A SURVEY OF INDUSTRIAL APPLICATIONS

ACKNOWLEDGEMENT

This is a slightly abridged version of the Forty-Fourth Thomas Hawksley Lecture delivered at a meeting of The Institution of Mechanical Engineers in London on 15 November 1957, and published in the May 1958 edition of "The Chartered Mechanical Engineer." The author is Dr. J. P. denHartog, Professor of Mechanical Engineering and Head of the Department of Mechanical Engineering, Massachusetts Institute of Technology.

INTRODUCTION

In this lecture an attempt will be made to present an outline of the historical development and of the present state of the art of vibration as they occur in the machines and structures of importance to mechanical engineering. This is but a small aspect of the general subject of vibration which indeed forms the foundation of the theory of light, much of nuclear atomic physics and physical chemistry, most of electrical and communications engineering, acoustics, and seismology. It is always difficult to set an exact starting date for any human activity, but if we leave out of consideration the pendulum clock and the early low-speed reciprocating steam engine, then our subject can be said to have begun at the turn of this century when the reciprocating steam engine became faster and more powerful, and the steam turbine made its appearance on the scene. From that day on, the field has grown exponentially like almost everything else in this our golden age of science and engineering, and it now so pervades our profession that hardly a piece of apparatus is built without some consideration of vibration entering into its design.

Unlike the case of heat engineering, for which the scientific theory was developed only after a century of successful operation of the steam engine, the scientific foundation to our subject was laid some 20 years before the appearance of the first practical applications. The year 1877 saw the publication of Lord Rayleigh's "Theory of Sound" [1] and in that same year Routh won the Adams Prize for an essay "On the Stability of a Given State of Motion." [2]

Routh's essay dealt with the phenomenon of hunting in a flyball governor of a steam engine, and it laid the mathematical foundation to the theory of dynamic stability, and to a class of vibrations called "self-excited." This subject has had an extraordinary broad development in the last two decades in connection with servomechanisms and automation. Again the principles laid down in Routh's paper, although not quite as all-embracing as those of Rayleigh's book, have stood the test of time and have found their principal technical applications many decades after the essay was written.

In the present lecture a rather large number of actual cases will be discussed which can be divided into two distinct groups: Forced Vibrations and Self-excited Vibrations.

A forced vibration is imposed on a system by an external alternating force which has an identity and frequency of its own, independent of the vibration of the system. Hence the system vibrates with the frequency of the external force and this "forced frequency" in general is different from any of the "natural frequencies" of the system. When the forced frequency coincides with a natural one, there is "resonance," a condition the designer usually attempts to avoid. From this it follows that the alternating force persists even when the vibratory motion of the system is prevented.

A self-excited vibration does not have an alternating force independent of the motion, but that force, sustaining the vibration, is a consequence of the vibratory motion itself. The classical example is the steam engine, where the alternating steam force on the piston is brought about by the alternating motion of the piston and the valve gear itself. The source of energy—the boiler steam—has no alternating properties, but acquires them by the motion only. As a consequence, most self-excited systems vibrate at one of their natural frequencies, a rule which is violated only when the self-exciting force is relatively very large, like in the steam engine or in a fluttering airplane wing.

When a vibration condition is encountered in a machine, the first and foremost action required is to diagnose whether the motion is forced or self-excited, and usually this can be done by measuring the frequency. The remedial process to be applied is completely different for the two possible outcomes of this investigation.

We now proceed to discuss forced vibrations in ships, Diesel engines, turbines and parts thereof, excited by centrifugal forces, ocean waves, nonuniformities in the flow of water or steam and the nonuniform torques caused by the intermittent combustion in Diesel engines.

FORCED VIBRATIONS

Balancing

Historically, one of the first manifestations of vibration in machinery was that caused by unbalanced inertia forces. At the turn of the century the principal rotating machines were large multi-cylinder steam engines and, when their speeds went as high as 100 rev/min, the piston and rod weights approached a ton and the stroke reached several feet, the inertia forces developed into a major nuisance. One of the first important solutions of a vibration problem was the theory of balancing such engines by placing counterweights on the crankshaft and by opposing the residual inertia forces of the various cylinders against each other. This is still important in the design of reciprocating gas or Diesel engines and is now almost universally practiced, although even today grossly unbalanced piston-type gas or air compressors with one or two cylinders are being built, and bought without protest by unwary customers. In the textile industry, looms are still being used and newly produced, which are unnecessarily badly balanced, and the speed at which such machines can be run is thereby severely restricted.

However, when we think of "balancing" in the present age, we rather consider machines such as turbine rotors, electric motors or ship propellers, which appear in perfect balance on the blueprint, and in which the unbalance exists only by accidental non-uniformities in the construction. Of course, this type of unbalance is several orders of magnitude smaller than the type referred to above in reciprocating machinery. With the advent of the steam turbine, more than half a century ago, the rotating speeds became so high that very accurate balancing became a necessity. Balancing machines were developed, which first were rather cumbersome contrivances with crude vibration indicators but which in the course of years became highly refined and extremely sensitive. A dramatic way of demonstrating the sensitivity of these machines is by calculating the distance δ by which the center of gravity of the rotor is displaced as a consequence of the application of the last correction weight in the balancing process. Such a correction applied to a steam turbine rotor of $W=20{,}000$ lb. on a standard balancing machine, used to its best available sensitivity, is 2 in. oz., so that in this case

$$W\delta = 0.125 \text{ in. lb.}$$

with a c.g. displacement $\delta = 6\times10^{-6}$ in. In this machine the bearings are of 9 in. diameter, with a customary oil-film clearance of 0.009 in., which is more than 1000 times as great as the c.g. displacement. The factory in question does not require this accuracy; the allowed tolerance is 10 times as coarse, demanding only a hundredth part of the oil-film thickness, but in practice the men in the shop balance "to the limit of the machine." If we wonder how such accuracy is attainable, it is useful to calculate the centrifugal force caused by this unbalance, which at the turbine speed of 3600 rev/min comes out to be 60 lb. The shop rules thus require that the rotor be balanced so that the centrifugal force at running speed be less than 600 lb., which is reasonable.

A simple formula shows these relations. Let us achieve balance so that the centrifugal force $wr\omega^2/g$ is reduced to 1 percent of the rotor weight W or:

$$wr\left(\frac{\pi}{30}\text{rev/min}\right)^2/g=0.01W$$

The displacement δ of the center of gravity is found from $W\delta=wr$, and substituting this in the above equation we find:

$$\delta=0.01g/\left(\frac{\pi}{30}\text{rev/min}\right)^2=350/(\text{rev/min})^2 \text{ in.}$$

Rev/min	100	1000	3000	10 000
δ, in.	3.5×10^{-2}	3.5×10^{-4}	4×10^{-5}	3.5×10^{-6}

We see that a standard 3000 rev/min turbine will jump on its bearings (centrifugal force equals weight) if its center of gravity shifts by 4×10^{-5} in. and such a turbine rotor operates with temperature differences between its various parts of several hundred degrees. In a good rotor these occur along the axis while the temperature is constant along any peripheral circle. If this is not the case, the center line will bend, throwing the machine out of balance. When a (horizontal) steam turbine rotor has stood still in its casing for some time, it is generally humped by the fact that its upper part is warmer than the bottom part; hence it is standard practice to turn the rotor over at slow speed for some time before putting it in operation.

A curious case of such "heat distortion vibration" was observed in a series of turbine rotors for a ship propulsion plant consisting of solid steel forgings in which the usual inspection hole was drilled along the center line. This hole being of 2 in. diameter and 6 ft. length, ending inside the forging at its deep end, could not, of course, be drilled exactly on the center line, and its far end was off center by about ⅛ in. This closed end of the central bore was at the high-temperature end of the turbine at some 600°F. By an odd set of circumstances about a half pint of cutting fluid (practically water) was left in this bore before it was sealed off by a screw plug at the open end. On account of the eccentricity of the bore, this water collected by centrifugal force on one side of it, leaving a part of the bore periphery dry. The high temperature boiled off the water in the bore, and the vapor condensed again some 5 ft. farther along the bore where the surrounding temperature was much lower. The boiling process on part of the periphery extracted heat from the rotor on one side causing it to bend, resulting in an intolerable vibration. The rev/min of the unit being approximately 5000, very little bending was required to bring this about. After the water was discovered and drained off, the machines performed satisfactorily.

Other cases of heat distortion, caused by rubbing, with the "hump" of distortion slowly wandering through the rotor have been reported in the literature.[3]

The fact that accurate balancing machines exist and are being used in many shops for all sorts of work leads to some strange inconsistencies when they are applied to slow-speed machinery. There is at least one shipyard in the world where it is customary to balance all propellers on one of the most sensitive large balancing machines commercially available. As an example, consider a large propeller with a rating of 25,000 h.p. at 100 rev/min. If this be corrected to 20 in. oz. (or better than that), the centrifugal force at service speed works out at 0.4 lb. The thrust force of this propeller at 20 knots is approximately 400,000 lb. or 100,000 lb. per blade for a four-bladed propeller. The thrust of each blade is proportional to its hydrodynamic angle of attack which is of the order of 3° or 4°. If one of the four blades has an angle of attack differing from the other three by 1 percent or by 0.04°, its thrust will be off by 1,000 lb., and this unbalanced hydrodynamic force, although parallel to the shaft, will have a moment arm about the last bearing about equal to the arm of the centrifugal unbalanced force of 0.4 lb. Thus the balancing machine in this instance is a fine toy, as harmless as a 10 decimal place calculating machine applied to physical quantities subject to 10 percent errors, or as innocent as a highly accurate vibration recorder and analyzer applied to a situation of which the physical significance is not clearly understood qualitatively.

Recently there has been an application of balancing which is considerably more accurate yet than the balancing for centrifugal force mentioned above. For inertial navigation systems small gyroscopes are used, of which the rotating wheel is enclosed in a sealed non-rotating can, which is partly floating in a viscous fluid and partly supported on a pair of jewel bearings, defining an axis perpendicular to that of the gyro spin. If the center of gravity of the can with its contents does not coincide with the jewel axis, the gravity force will exert a moment or torque about it, and for proper operation this must not be the case. Hence it is necessary by applying correction weights to the outside of the can to bring its center of gravity very close to the jewel axis. This is done by rotating the gyro at its service speed inside the can, subjecting the assembly to the influence of gravity for several hours and measuring the very slow precession resulting from the gravity torque. This precession is then reduced to a minimum by proper balancing in a number of trial runs, and the accuracy of the process can be described, rather startlingly, by the displacement of the center of gravity of the can caused by the last balancing correction. This is 0.6×10^{-6} in., or some 40 times smaller than the wavelength of visible light, or about 1,000 times smaller than the best possible geometrical definition one could give of the center of a jewel bearing. In contrast be it men-

tioned that, in thousands of textile looms all over the world, the center of gravity of the "lay" or principal reciprocating part is from 8 in. to 1 ft. from its axis of rotation.

Longitudinal Vibration of Ship's Shafting

A ship's propeller running in open undisturbed water has a perfectly smooth torque and thrust, but any actual propeller is mounted sufficiently close to the hull or to appendages like struts or the rudder stem so that the velocity distribution of the water it meets is not uniform over the propeller disc. This means that an individual propeller blade will experience time-variable hydrodynamic forces with the result that the torque as well as the thrust of the entire propeller will have a reasonably strong component of blade frequency (number of blades times rev/min) and weaker components of multiples of that frequency. The torque variation may cause torsional vibration in the shaft system and the thrust variation may cause longitudinal vibration. A shaft is reasonably soft against torsion, but it is very stiff against longitudinal tension or compression. Hence the longitudinal natural frequency of a shaft-propeller system is high, so high that in practically all ships it is above the exciting blade frequency. Only in the very largest ships having shafts several hundred feet in length and running at relatively high propeller speed (200 rev/min) have longitudinal resonances been observed. On the other hand, torsional resonances are quite common. The torsional and longitudinal systems are almost alike physically and mathematically. In torsion the moment of inertia of the propeller I_{prop} works against the torsional stiffness GI_p/L of the shaft, and there is a node near the main gear. In the longitudinal case the mass of the propeller M_{prop} plays against the tension stiffness EA/L of the shaft, again with a node near the main gear. The natural frequencies are roughly

$$\omega^2_{tors} = GI_p/LI_{prop} \text{ and } \omega^2_{long} = EA/LM_{prop}$$

The ratio of these frequencies is:

$$\frac{\omega^2_{long}}{\omega^2_{tors}} = \frac{E}{G}\left(\frac{A}{I_p}\right)_{shaft} \cdot \left(\frac{I}{M}\right)_{prop} \approx 2\tfrac{1}{2}\left(\frac{\rho_{prop}}{\rho_{shaft}}\right)^2$$

so that

$$\omega_{long}/\omega_{tors} = 1.6\rho_{prop}/\rho_{shaft}$$

where ρ is the radius of gyration, about 70 percent of the outer radius, both for the shaft and for the propeller. It is seen that the longitudinal frequency exceeds the torsional one by more than the ratio of the propeller diameter to the shaft diameter, roughly 10 to 1. Hence longitudinal and torsional trouble is never encountered together on the same ship. The torsional trouble occurs in tankers with short shafts, where the longitudinal would be five or more times the running speed; longitudinal resonance has been observed only in battleships and superliners with very long shafts, where the torsional critical speed is below 20 rev/min. Such a longitudinal resonance, at blade frequency, about 10 c/s, can be (and has been observed to be) extremely violent: the thrust variation in the propeller is of the order of 10 percent of mean, fluctuating between 90 and 110 percent.[4] At the critical speed this is magnified by a resonance factor of more than 10, so that the actual thrust on the ship's structure varies between a small rearward pull and more than double the mean. In the usual construction this enters the ship's structure at the thrust bearing in the engine room about midships, and as a result the amplitudes in the engine room, especially of piping and the spring supported condenser, become frighteningly large. The vibration penetrates into the remotest parts of the ship and wherever there is a local element with a natural frequency of the order of the critical, it responds.

The remedies for such a condition in a new design consist in a proper choice of the number of propeller blades, and a proper location of the thrust bearing along the shaft, farther aft than the engine room. With these two variables the critical speed can be shifted, by precalculation, into a position where it can be tolerated, either above the top rev/min or at a low rev/min where the thrust is small.[5]

By properly designing the shape of the propeller, and by increasing the clearance between its blade tips and the hull, the thrust variation can be brought down to a smaller value. The calculation of this effect is not very reliable, but it can be determined experimentally by model tests in a towing tank.[4]

In an existing ship with vibration trouble of this type, the only practical possibility is the use of a propeller with a different number of blades. For almost a century now ship's propellers have been standardized to have almost exclusively either three or four blades, and the introduction of five-bladed ones in 1941 for this purpose was looked upon with some scepticism. However, these doubts have now largely been dissipated by the experience on many ships and by exhaustive measurements on models in propeller tunnels. The five-bladed propeller, properly designed, appears to have as good a propulsive efficiency as the more conservative three- or four-bladed, and it is substantially smoother. The actual percentage of torque or thrust variation of a propeller depends not only on its own design but on its vicinity to the hull and in general on all the complications of the flow distribution in its disc, so that no definite statements can be made on a propeller design by itself. However, there are some important general tendencies. In the first place, if the propeller could be built with an infinite number of blades, it would be entirely smooth. Since the word "infinite" is synonymous with "large," we suspect that a five-bladed propeller would be smoother than a four-bladed and much smoother than a three-bladed. In general this has been found to be the case: the five-bladed on the average produces only half the disturbance of a three-bladed propeller.

However, the four-bladed one does not quite fit into this picture because the interference of the ship's hull with the flow often consists of a long vertical member like a rudder stem, and in a four-bladed propeller two blades come into its wake simultaneously. This makes the four-bladed and in general any propeller having an even number of blades rougher than the nearby odd-numbered one. Hence it is expected that a seven-bladed propeller would come out very smooth indeed. As far as we know, it has never been tried, and it is quite possible that its propulsive efficiency would be unsatisfactory.

The disturbances caused by the small number of propeller blades are not limited to the forces by the water on the propeller, resulting in torque and thrust variations. The pressure field of the rotating propeller blade extends a considerable distance beyond tips and consequently there is a periodic pressure variation (at blade frequency) exerted by the water directly on the hull. These forces make the stern of any high powered vessel rough, irrespective of resonance. In this connection the five-bladed propeller again proves to be superior to three- or four-bladed ones, and a seven-bladed might have a distinct advantage.

Anti-roll Devices for Ships

Seasickness is decidedly unpleasant and attempts to attack it at its source by mitigating the roll of ships in a wavy sea have been numerous and varied. There is a large bibliography of the subject in Chadwick.[6] Historically the first of these attempts consisted in the fitting of a pair of "bilge keels," which are submerged fins, extending along a substantial part of the length of the ship, protruding perpendicularly from the hull by not more than a foot. The rolling of the ship is impeded by hydrodynamic damping of these keels; it is a simple method, but unfortunately not a very effective one.

Two other early schemes, both before the turn of the century in Britain, are due to Thornycroft and to Bessemer. In the first one, a heavy weight was mounted eccentrically rotatable about a vertical axis midships; the axis could be turned 90° to each side by a steam piston, thus moving the weight from side to side on the ship. The rolling moment of the weight on the ship could thus be varied, and the valve of the steam piston was controlled by the roll angle of the ship, like in a modern servomechanism. This construction was 50 years ahead of its time; it was recognized much later that the moment (weight times stroke) required for proper operation has to be very large, certainly very much larger than in Thornycroft's design. Bessemer utilized a gyroscope, which was mounted solidly in the ship, so that it could not "precess." The effect of this is to couple the rolling and pitching motions of the ship. Since usually the rolling motion is more obnoxious than the pitching one, this construction conceivably might diminish the roll at the expense of increased pitch, although that is doubtful.

The numerous constructions that have been built during the last 40 years can be divided into three groups: (1) those in which a weight (or water) moves from side to side; (2) those utilizing large precessing gyroscopes; and (3) those using the hydrodynamic action of submerged fins. Each one of these systems is said to be "passive" when it is not controlled by some sensing instrument and a servomechanism; it is said to be "active" when such a control is applied.

The moving weight systems were used extensively on large ships from 1910 until recently in the form of two partially filled water tanks, one on each side of the ship, with a water connection between them, which at first took the form of a pipe inside the hull and later became an open connection through the sea outside the hull. In the latter case there is an inboard air-filler pipe connecting the tops of the tanks. These constructions, installed on several German passenger liners, are due to Frahm,[7] and the tanks are commonly called Frahm tanks. When the ship rolls, water tends to flow from one tank to the other, and this motion is partly impeded by a throttle valve in the connecting pipe (at first in the water pipe, later in the air pipe). A certain amount of energy is destroyed at the throttle valve and in the pipe. For a wide open throttle this energy loss is very small (much flow, no pressure drop), and again for a completely closed valve it is small (no flow, large pressure drop). For an intermediate setting of the valve, when there is some flow and some pressure drop, the energy loss attains a maximum. The only source of this energy is the rolling motion of the ship, so that the system introduces damping into that motion. Since no servomechanism is used, this system is called a passive one. In an active water tank system the flow of water is brought about by a pump, the action of which in turn is controlled by some sensing mechanism, usually a small gyroscope, which feels the rolling motion of the ship. Such active tank systems have been built for one or two destroyers of the United States Navy.[8]

The gyroscopic construction was originated by Schlick in Germany, the same engineer who deduced the rules of inertia balance in reciprocating multi-cylinder engines. A very large gyro wheel rotates about a vertical axis and its upper and lower bearings are mounted in a gimbal or frame which itself carries two trunnions enabling it to pivot about an athwartships axis. When the ship rolls, the gyroscope tends to precess about this athwartships axis, executing a pitching motion inside the ship. This pitching motion is impeded by a brake. Again when the brake shoes are loose, no energy is dissipated (plenty of motion, no force); when the brake shoes are completely tight, no energy is lost either (no motion, plenty of force). The brake actually is set to permit some slipping at a considerable torque in order to kill a maximum amount of

energy, which is extracted from the rolling motion, thus constituting a damping. This system is passive; it could have been made active by controlling the brake shoe pressure by means of a small gyroscope "sensor." This, as far as we know, was never done; instead of controlling the brake, it was replaced by an electric motor (the "precession motor") with a slow-speed worm-gear drive, and the speed of this precession motor was varied by the control. It is clear from the foregoing, however, that the precession motor does not really operate as a motor, because it never pushes the precessing gyro, but on the contrary is always pushed by it. This system is due to Sperry in the United States, and it was applied to numerous ships from 1915 on until recently.

The anti-roll device employing submerged fins in its passive form is the pair of bilge keels mentioned previously. When the ship is rolling without forward speed, these keels only stir the water somewhat, leaving turbulent eddies above and below them; but since the rolling motion is slow, there is very little energy dissipated this way. When the ship is making headway and is rolling simultaneously, a new effect comes in. The bilge keel can be looked upon as a wing under water, aligned in the direction of the relative water flow when the ship is going forward. When there is no roll and only forward motion, the water does not push the bilge keel up or down, the lift and the angle of attack both being zero. When a roll is superimposed on the forward motion, the relative speed of the water has a small vertical component proportional to the rolling velocity, and this causes an apparent angle of attack with a consequent lift or downpush on the keel. The direction of this force is opposite for the two keels on each side of the ship, and the direction of the force also is opposite to that of the vertical velocity of each keel, thus constituting a damping of the rolling motion of the ship. This effect of the bilge keels is more or less proportional to the square of the forward speed.

Finally, the activated form of the hydrofoil system consists of having two submerged fins protrude perpendicularly from the hull, which now have a greater resemblance to airplane wings than the bilge keels, by having a better length/chord ratio or "aspect ratio." If these fins were fixed to the hull, they would operate like improved bilge keels, but now they are built pivotable about a horizontal athwartships axis and their angle of attack is changed periodically through a few degrees by a motor inside the ship, which in turn is controlled by a small gyroscopic sensor.

The usual control is to set the instantaneous angle of attack of the fins proportional to the velocity of the ship's roll. This is the Denny-Brown system,[9] which is currently being applied to most major passenger ships and which has displaced all other systems. Apart from its inherent simplicity, the principal reason for this is that it is capable of exerting much larger forces on the ship than its competitors.

With the other two systems, completely satisfactory operation never was achieved because for proper performance the weight devoted to the tanks or gyroscopes should amount to some 5 percent of the total tonnage, and this was never done. It is to be noted that the fin system depends on the forward speed and does not operate at standstill, while the other systems do. Ships, however, do not habitually lie still in a heavy sea.

Torsional Vibrations

These often occur in the shafting systems of power machinery when the torque either of the driving or of the driven machine shows periodic variations in intensity. Torsional vibration problems and their practical solution form the subject of a work by Ker Wilson.[10]

Troubles and failures of shafting probably occurred first, shortly after the turn of the century, in ships driven by triple-expansion steam engines, but the problem was not recognized and explained until the 1914-18 war when the Germans encountered an epidemic of shaft failures in their Diesel-driven submarines.[11] Shortly after, about 1925, methods of calculating and of designing away from torsional critical speeds became generally known.[12] With increased powers and longer shafts in Diesel-driven ships, it soon became the better part of wisdom to precalculate all such drives. That the problem actually existed before it was recognized is indicated by the fact that during the serious shipping emergency of 1942 a rather large number of simple "Liberty" ships were built in the United States, powered by triple-expansion steam engines, from drawings of a successful design made in 1916, dusted off for the occasion, but not checked for torsionals. These ships served their purpose adequately during the war, but afterwards, from 1946 on, started to drop their propellers into the ocean. It was found that when after the war the necessity for convoy sailing had disappeared, commercial competition induced the operators to raise the rev/min somewhat, bringing the ships into their principal engine-excited torsional critical speed, which fatigued the main shaft. The design was of the 1914-18 war period, conservative in that the steady full-load stress in the shaft was no more than 15 percent of the yield stress, although the keyway at the propellers was cut with sharp corners.

From the standpoint of torsional vibration the Diesel engine is the worst existing engine, because it has the greatest non-uniformity of torque: in a single cylinder the intensity of the alternating torque at firing frequency is several times as great as its mean value. However, torsional vibration has been and is being caused by other less offensive machinery. The geared-steam-turbine ship drive has an almost ideally smooth driving torque, and

then the driven propeller with its small number of blades becomes the exciter, because in the usual installations its alternating torque at blade frequency is of the order of 10 percent of the steady torque.[4] This generally is not sufficient to break the shafting, although even that has happened on occasion, but at a critical speed it can feed considerable alternating torque into the main gear, which is extremely sensitive to such abuse. Other torsional excitations of less frequent occurrence are in electric motors or generators with 100 or 200 c/s frequency of the torque (which goes to zero twice during each current cycle) or in squirrel-cage type induction motors with the slip frequency of the squirrel-cage bars. It has now become general practice to precalculate torsional frequencies in all machinery where non-uniform torques appear.

In fact, we sometimes do better than that; government specifications have been written demanding natural torsional frequency calculations in installations where no conceivable excitation exists, so that nobody would know what to do with the answers after they are produced because nobody knows what frequency should be avoided. Thus, we have indeed gone far since 1877 when Lord Rayleigh wrote on the subject. He was lucid and concise as usual, saying (Vol. 1, Chapter 7 on "Longitudinal and Torsional Vibrations of Bars," Section 159): "The second class of vibrations, called torsional, which depend on the resistance opposed to twisting, is of very small importance."

The principal and best method of producing a good torsional design is to place the critical speeds at some distance, say, 15 percent or more, from the operating speeds by suitable dimensioning of the shaft diameters and flywheel effects. This is always possible and convenient when we deal with a constant-speed machine, but if operation in a wide speed range is required, it is more difficult and sometimes it becomes practically impossible. Then we are obliged to use dampers or tuned vibration absorbers of the centrifugal pendulum type. The latter device was invented in the early thirties and was of great importance particularly in aircraft piston engines of which they became and in many instances still are an essential element.[13] After 1945 they were applied to land and ship Diesel engines as well, but now they have more or less had their day and have given way to plain unturned friction dampers. This damper in its fundamental form was invented by F. W. Lanchester of automobile and aeronautics fame and consists of a small flywheel coupled to the shaft by friction only, either dry friction or viscous fluid friction. A damper used to be especially designed and built for each engine or engine application, but for the last decade they have been available on the market ready made in different sizes. This modern commercial form of the Lanchester damper consists of a loose flywheel completely enclosed in a housing with small clearance, the clearance space being entirely filled with a viscous silicone fluid.

In Diesel ship drives the propeller in the water produces a certain amount of damping, particularly when its moment of inertia is small with respect to that of the engine with its flywheel, so that the propeller then has a relatively large amplitude of vibration. However, in cases where a Diesel is coupled to an electric generator there appears to be no source of damping other than that in the engine itself and for the past 30 years attempts have been made to calculate this damping, with little result. A reason for this lack of success dawned on the lecturer one day while he was standing in the forward anchor-chain locker of an oil tanker under way. The ship had a Diesel drive and was running in one of the torsional criticals, of which the amplitude had been calculated based on propeller damping and the usual engine damping. However, a torsiograph test revealed only about half the calculated amplitude, so that there was a hidden source of damping somewhere. It happened in this case that the frequency of the torsional coincided with one of the natural frequencies of the ship as a beam in bending. Now a straight shaft in torsional vibration imposes no alternating force on its bearings, but a crankshaft does, so that in an engine there is a coupling between the torsional crankshaft motion and a lateral or vertical motion of the main bearings. Thus the whole ship took part in the vibration and one of the links of the anchor chain lying on the steel floor 400 ft. from the engine happened to resonate and tapped out the torsional critical frequency audibly, thus dissipating some energy and diminishing the torsional amplitude of the engine. Hence, we have now generally come to the conclusion that the damping in an engine is so complicated that its successful calculation is impossible, and we go by the empirical rule that at a critical speed in the absence of dampers or propellers the maxium alternating torque in the shafting is roughly 50 times the exciting torque of the engine.[10] The factor 50 varies between 25 when we are lucky with rattling anchor chains and 100 when we are unlucky and shaft failures occur that we did not quite predict.

Turbine Blades

The blades of steam and gas turbines or compressors operate in considerable centrifugal fields, often at high temperatures, and are required to do this without failure for thousands of hours. Their vibration frequencies are of the order of hundreds per second, so that the life of a blade is measured in thousands of millions of cycles. Ever since the introduction of the steam turbine more than half a century ago, this problem has been with us, and although much has been learned and the blades can be predesigned to great reliability, the problem keeps cropping up in new modifications and occasional failures from this cause continue to plague us.

At the outset it is remarked that blade failures have occurred from two fundamentally different causes: (1) forced vibrations brought about by circumferential non-uniformities in the steam or gas flow and (2) self-excited vibrations from "stall flutter," to which we will return later. The stall flutter trouble is limited in practice to the blades of air compressors of aviation jet engines, the steam turbine being free from it.

An individual rotating blade during its revolution passes a row of stationary guide vanes or nozzles from which the steam issues. It is obvious that just behind each guide vane the steam velocity is less than in the middle of each passage between two guide vanes, so that the blade in question is subjected to a series of impulses of guide vane frequency. For reasons of proper flow guidance the guide vanes are almost always equally spaced, so that these impulses come regularly, at a high frequency, for example, 50 guide vanes at 3000 rev/min = 2500 c/s for a steam turbine or 50 guide vanes at 12,000 rev/min = 10,000 c/s for an aviation gas turbine. These frequencies are invariably very much higher than the lowest natural frequency of the blade (or interconnected group of blades), but resonance may occur with one of the higher natural modes of motion of a blade. The higher the mode of motion of a blade and consequently the more complicated its geometry of distortion with many nodes, the more difficult it is to excite such a mode, but on the other hand, the intensity of the excitation at guide vane frequency is also high and failures have occurred which are explained by this mechanism. They are relatively rare and not too serious, because they usually manifest themselves by the break-off of a small tip of the blade rather than a failure at the root, which is more dangerous.

Apart from the very pronounced guide vane or nozzle harmonic in the flow distribution, there are other non-uniformities of one, two, three, etc., spots of maximum velocity around the periphery. These "low harmonics of the velocity distribution field" are of extremely low intensity and are caused by unavoidable inaccuracies in the construction. They should be zero, just like a cylindrical rotor should be in perfect inertia balance, but they are not zero for about the same reason. Excitations of such low intensity can usually be ignored, but not in this case. In the first place they are apt to resonate with the first few low modes of motion of the blade or group, which are very easily excited, and in the second place a turbine blade at high speed is an element with extremely low damping, which at resonance can magnify an excitation a hundredfold or more. In other words if the steam pressure distribution were to be 1 percent larger than the mean in two diametrically opposite points and 1 percent below normal at two points 90° in between, and if the blade rotated through this at the appropriate speed to be "in resonance with this second harmonic of the field distribution" then the alternating stress in the blade would be equal to that caused by the full steam pressure, and this in combination with the steady centrifugal and steam pressure stress might well fatigue the blade to destruction. Therefore, designers precalculate the blade frequencies and also determine them by laboratory test and then modify them to avoid such resonances wherever possible. If they cannot be avoided, as in turbines required to operate over a wide speed range, the blade design for steady stress has to be made very conservative.

A pronounced non-uniformity in the circumferential steam velocity pattern is encountered when "partial admission" is used. The quantity of steam which in a dense, high-pressure state passes through the first row of blades is the same which, greatly rarefied, flows through the last row into the condenser. This means that the cross-sectional area for steam passage at the boiler end must be held small and a design in which the steam is admitted all around the 360° might give embarrassingly short blades. Rather than do this with blades ⅛ in. long, which means inefficiency and loss, the steam is admitted over part of the periphery only, say, over 20°, so that the blades can be made longer for the same steam cross-section. An individual rotating blade then runs through 340° without steam, then suddenly gets hit with the full load and 20° later receives an equally sudden shock by the removal of the load. This is very rough treatment and is possible at all only because it is applied to the first row of blades which are very stiff and rugged. An epidemic of failures of such blades occurred in the 1930's for which no explanation was available because the stresses imposed by the sudden loads were quite conservative.[14] Experiments measuring the tip motion of the blades in the turbine under load showed that after the shock the blade vibrated, which was expected. The vibration at the natural frequency was at about 50 to 100 cycles per revolution of the turbine.

The surprising fact was found that the damping of the blade was so small that while performing more than 50 cycles during its 340° vacation, the blade ended up with a vibratory amplitude scarcely less than when it started. In fact, it would take almost a thousand cycles to make it die down, like a church bell. Then the blade still vibrating re-entered the steam jet and received another pair of shocks. Statistically some of the blades would receive these repeated shocks so that their resultant vibrations were in the right phase to become additive and after a few seconds the amplitude would be 20 times as large as the one caused by a single shock. That trouble was overcome by various means; by designing the edge of the nozzle such that the square edges of the steam jet were rounded off,[14] by more conservative static design and primarily by replacing the blade material by a special

alloy steel, of which the internal damping at the high temperature involved was several times higher than that of the carbon steel used previously. This new alloy has been used extensively in steam-turbine blading since that time.

Steam turbines for marine use and gas turbines for aircraft propulsion have to operate in a wide speed range, and for such applications it is of extreme importance to find a means of introducing damping into the blades, so that they become capable of accepting the unavoidable excitations without excessive amplitude and stress. While this has been universally realized for several decades and dozens of patents have been issued, no really good solution has been found so far. Steels or other metals dissipate energy into heat when subjected to alternating stress, and the amount so dissipated per cycle grows with the intensity of the stress, slowly at first and very steeply later. Unfortunately in all known metals the stress at which the damping becomes large is also the fatigue stress at which failure occurs; in fact the appearance of large damping is often considered to be an indication of incipient failure. A very good way of dissipating energy is to clamp the vibrating body loosely; small relative motions at the clamp cause friction with large dissipation. This fact was not well understood for a long time; most figures for the energy dissipation in metals (known also as "mechanical hysteresis loss") published between 1900 and 1940 are several times larger than the truth. The experimenters included the clamping loss in their measurement and did not realize its relative importance. A turbine blade is clamped at its root and when the damping is measured in the shop, it is gratifyingly large, but unfortunately, when the blade is running at full speed, the centrifugal force is so large that it freezes the connection solid and the "root damping" becomes negligible. This is the case for the properly clamped blades of steam turbines as well as for the loosely mounted ones of aviation gas turbines. Constructions have been proposed and tested in which loose rods, wires, balls, shot, or a liquid are inserted inside the hollow blade, with the object of obtaining friction dissipation by relative motion, but it seems that the centrifugal force freezes them all. Nevertheless, since the reward for a satisfactory solution is great, the search goes on, by the mechanical engineer as well as by the metallurgist, and we may look forward to success in the future.

Calculations of Natural Frequencies

Usually the first and most important step in the calculation of a vibrating system (after the phenomenon is understood physically in a qualitative way) is the determination of its natural frequencies.

Many methods of doing this exist, almost all of them inevitably disclosed in principle in Rayleigh's book, and later rediscovered and elaborated by other authors. In this situation a revolution has been brought about during the last decade by the advent of the electronic computer. Before the computer age a tacit but most important consideration was the length of time it would take a man to perform a calculation and, as a result, the system was idealized or simplified very thoroughly and then approximate methods of calculation were employed, simply because the more exact ones took too much time. This condition has now totally changed and an engineer who has the use of a computer no longer attempts to simplify his system, but on the contrary will put in all complications which even remotely might affect the result. He will also direct the computer to calculate his system for a number of design variations, because once it is set up that calculation is made very quickly.

However, on the general principle that one never gets anything for nothing, a new difficulty has arisen: that of "programming" the computer operations for the problem in hand. Therefore, the method of calculation chosen is no longer the one which takes the shortest time by hand, but rather the one which can be most conveniently "programmed" and which then gives the maximum information.

This has brought to the fore a method of computation for flexural beam problems, such as rotors or turbine blades, which was originally used for manual computation in torsional systems only. It is there known as the "Holzer" method,[11] and it consists of imposing on the system a forced oscillation of some frequency, starting with a given amplitude (usually unit amplitude) at one end and then finding what excitation (torque or force) is required at the other end to bring this about. At a natural frequency the torque or force required for its maintenance reduces to zero.

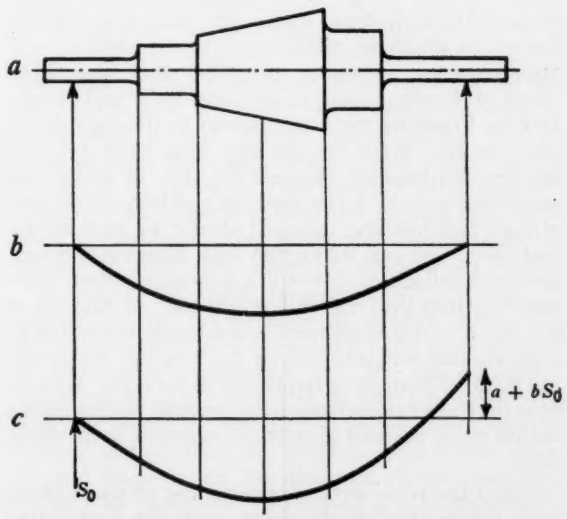


Figure 1. The Holzer Method Applied to the Calculation of Natural Frequencies of a Beam in Bending.

For a torsional system we thus start by making the best guess we can for the natural frequency and we then impose a forced torsional vibration of unit angular amplitude at that guessed frequency on one free end of the engine system. The consequent required inertia torque, shaft torque, shaft wind-up and torsional angle are then calculated step by step along the system until we find a certain torque at the other free end which has to be imposed on that end from the outside in order to make the system vibrate as calculated. If the guessed frequency happens to be a natural frequency of the system, this necessary torque or "remainder torque" becomes zero, because the system can vibrate freely at the natural frequency without a forcing torque from the outside. The method consists in making a number of trials with different frequencies until the remainder torque becomes sufficiently small. This calculation by slide rule is greatly facilitated by a convenient tabular form in which the operations are recorded—known as the Holzer table. This procedure has been in use continuously since 1922.

The first application to a more complicated case involving computers was in connection with the bending critical speed of rotors on two or more bearings,[15, 16] Consider the simplest case of a rotor on two supports, Figure 1a, vibrating in the shape of Figure 1b at an unknown frequency. If we make a rough guess for that frequency, we can start at the left-hand end, Figure 1c, with a slope of one unit and an unknown bearing force or shear force S_0. We can now calculate the curve step by step along the rotor, applying classical beam formulae to each short section of it. At the left-hand end of such a section the deflection, slope, moment, and shear are known from the previous step, and the vibrating inertia load on the section is calculated from the assumed frequency and the amplitude. Then by beam theory the corresponding four quantities at the right-hand end of the section are determined. In all these calculations the symbol S_0 will appear as an unknown quantity, since the sheer force or bearing load at the left-hand end is unknown. Pursuing the computation to the right-hand end, we find there for the end amplitude $a + bS_0$, and for the bending moment $c + dS_0$, in which the quantities a, b, c, d are definite numbers. Both amplitude and bending moment should be zero at the end. We now can make the end bending moment zero and deduce $S_0 = -c/d$, a known number. Substituting that into the end amplitude, we find for it $a - bc/d$, which in general is a number not equal to zero, coming out zero only if the original choice for the natural frequency happened to be right. We can now perform calculations for a number of frequency values until the end amplitude becomes sufficiently small.

When the rotor extends over three or more bearings, the method is the same until the first intermediate bearing is reached. Then the shear at the left bearing S_0 is calculated from the condition that the amplitude at the intermediate bearing is zero. But the intermediate bearing force itself then enters as a new unknown S_1, and with that we proceed to the next intermediate bearing, and so on.

The method is equally applicable to other "beams" with different end supports. For a cantilever turbine blade, for example, we can start at the root end with zero deflection and slope, with a bending moment equal to unity and an unknown shear force S_0. Proceeding by steps to the free end of the blade, we find expressions containing the letter S_0 for the end bending moment and the end shear force. Making one of these, say, the shear force, zero, calculating S_0 and substituting gives us an end bending moment which should come out to be zero if the original choice for the frequency were correct. A number of such computations with different assumed values for the frequency leads to a number of different "remainder bending moments" and, by interpolation or plotting, the true natural frequency with zero remainder moment can be found. In a single isolated turbine blade the cross-section varies along its length; the centers of gravity of the sections do not generally lie along a straight radial line, so that the centrifugal force has a variable bite on it; the section twists or spirals from bottom to top; and for the higher or multi-modes of motion, it becomes necessary to consider the effect of shear forces on the deflection. This assembly of complications makes the calculation enormously time consuming and hence amenable only to a high-speed computer. When the blades occur in groups connected together with lashing wires or a shroud, the problem can be treated by matching end conditions at the joints, in a manner similar to that just described for multi-span rotors.[16]

For turbine discs, jet engine housings, and other elements, an idealization in terms of "beams" is no longer adequate and "plate" theory has to be brought in. For these the relaxation method of calculation, which was the subject of the 1953 Thomas Hawksley Lecture by Sir Richard Southwell has been found most useful.[17] At present this method does not seem conveniently amenable to computer programming, and it remains to be seen which of the existing methods or which new method will ultimately be found most suitable for determining natural frequencies of plate and shell-like structures by means of the digital computer.

SELF-EXCITED VIBRATIONS

Vibration Excited by Dry Friction

A general class of vibrations exists in which the self-excitation is brought about by dry friction forces, which have the peculiarity of being greater at rest than in motion. Familiar examples are the screeching of unlubricated rubbing parts or the mechanism of a violin, which indeed requires the application of rosin on the bow to accentuate the

difference between the friction forces at rest and in motion. The phenomenon was known and practically used for centuries before its scientific explanation, which in Rayleigh's words ("Theory of Sound," vol. I, Section 138) is: "The sustaining power of the violin bow depends upon the fact that solid friction is less at moderate than at small velocities, so that when the part of the string acted upon is moving with the bow (not improbably at the same velocity), the mutual action is greater than when the string is moving in the opposite direction with a greater relative velocity. The accelerating effect in the first part of the motion is thus not entirely neutralized by the subsequent retardation, and an outstanding acceleration remains capable of maintaining the vibration in spite of other losses of energy."

This type of motion often is described as a "chatter" or a "stick-slip vibration," and although it occurs more often than it should, it will usually respond to a drop of oil, judiciously and timely applied. There is a case on record in which a rather large bascule-type bridge failed by structural fatigue as a result of vibrations occurring during the raising and lowering operations over a period of months, caused by a bearing from which the grease pot was spared for too long a time.[18] The only places where this type of disturbance assumes technical importance are where lubricants either are not wanted or are ineffective, some examples being (1) automobile clutches, (2) machine tools cutting into metal, and (3) outboard, rubber-lined, water-lubricated, shaft bearings.

In the first two of these, where the unfavorable friction characteristic is often unavoidable, the effects are kept within bounds by stiffening the construction, thus reducing the ratio between the exciting friction force and the elastic restoring force for a given amplitude of motion.[19, 20] Dampers in the form of inserts in the clamps of machine tools have been tried, and show some promise, but their practical merits at this time are doubtful. The water lubrication in the rubber-lined outboard bearings of ship's propeller shafts is reasonably effective at most service speed, but does not work at extremely low speeds, which can result in a high-pitched squealing noise just after starting from standstill. This noise, broadcast through the water by the propeller blades, which act like a loudspeaker, was not appreciated during the war by the crews of submarines trying to get away undetected.

Oil Whip

In a film-lubricated journal bearing the angular direction of the load does not coincide with the direction of the minimum thickness of the oil film, as is shown by theory as well as by experimental observation. In normal operation, when the load on the usual horizontal bearing is the weight vertically downwards, this means that the point of minimum oil-film thickness is found angularly ahead of the downward vertical in the direction of journal rotation by an angle somewhat less than 90°. Now consider a machine with a vertical shaft in which the journal bearings are guides only and normally carry no load, and imagine this machine to be whirling in a slightly bent state at its natural frequency. Then the geometry of the motion requires the thin spot of the oil film to rotate and be angularly in line with the rotating centrifugal force. The force of the oil film then, however, is not in the direction of the centrifugal bearing load, so that no equilibrium exists, and the component of the bearing pressure force perpendicular to the centrifugal force or eccentricity tends to excite the whirl. For this mechanism a well-developed oil film is essential and this is brought about by the shaft rotation. If in the absence of shaft rotation an elastic whirl is initiated, no oil film is developed and the whirl is soon damped out. In general for slow rotation and fast whirl the system is stable and conversely for fast rotation and slow whirl (that is, low natural frequency) the system is unstable and the whirl is maintained. Satisfactory experiments, supported by somewhat doubtful theories, place the neutral point (for a full 360° bearing) at a running speed equal to about twice the critical speed, which means that a full 360° vertical bearing rotating faster than twice its critical speed is inoperable. This rule is substantially correct for normal weight-loaded horizontal bearings as well, and the trouble increases in intensity with increasing speeds over and above twice the first critical. In the early days of the Parsons turbine for ships as well as for electric generation the rotors were made stiff, to run below the first critical; the transition to running speeds higher than critical came later and it was not until the early 1920's that the speeds reached twice the critical and the phenomenon of oil whip was discovered by Newkirk.[24] The generally accepted remedy is to relieve certain portions of the bearing by giving them substantial clearance, thus breaking up the 360° circle into a number of shorter arcs, none of which individually should exceed 135°. Several theories have been proposed for this, which do not appear to be quite as convincing as the practical experimental evidence. Bearings of this type have been in continuous use in the steam-turbine industry for 30 years with generally satisfactory results.

However, the trend towards still higher speeds (relative to the lowest whirling critical) is continuing and every year it becomes a little more difficult to get by with partial arcs only. For special experimental machines, running at 10,000 rpm, some firms now are using pivoted-pad bearings. The great invention of the tilted- and pivoted-pad bearing, based on and inspired by the theories of Osborne Reynolds was made at the turn of the century independently by Mitchell of Australia and Kingsbury of the United States. These pads have been applied

universally to thrust bearings, under the name of either of the inventors, but their introduction on journal bearings, where the pads are segments of a cylinder instead of flat plats, has been sporadic and tardy. These pivoted pads are completely stable as far as oil whip is concerned, and it is quite possible that they will find more widespread application in high-speed machines, in spite of their unquestioned greater cost and complication.

Aircraft jet engines generally run well above twice their first critical, and the fact that they almost all use ball bearings instead of film-lubricated bearings is possibly explained by the fear of oil whip.

Recent very interesting experiments by Hull, soon to be presented to the Institution's sister society in New York [22] are shedding new light on this question.

Leakage Through Narrow Passages

A group of self-excited phenomena has been observed in machines of which certain parts (sometimes large parts and sometimes small ones) vibrate at their natural frequency and by so doing periodically change the width of a narrow passage (for example, a labyrinth seal) through which a liquid or gas is flowing. The periodic changes in passage width cause velocity and pressure fluctuations of the fluid, which, when integrated over some area, cause periodic force variations on the vibrating machine part, obviously of the same frequency as the motion itself. When the phase relation is such that the excess pressure force on the vibrating part acts in the same direction as the vibratory velocity of that part, the fluid puts work into the vibration, which thus becomes self-excited.

A musical instrument based on this principle is the old-fashioned bellows-operated reed organ. The vibratory part there is the reed, being a thin steel cantilever about ½ in. long and rather less than ⅛ in. wide, which fits into a rectangular hole of the same size in a wall separating the bellows-fed pressure chamber from the open atmosphere. When the reed vibrates at its own natural frequency, it periodically varies the width of the passage; the air issues in puffs and the consequent pressure variations push the reed more in the direction of its own instantaneous velocity than against it, thus keeping it vibrating. The details of the flow and of the pressure variation are difficult to follow by theory.

In steam as well as in hydraulic turbines the fluid flows from a non-rotating high-pressure vessel into a non-rotating low-pressure vessel. Practically all of the fluid does that passing through the vanes or blades of the intervening rotating element, as intended, but since the rotating element cannot be attached to the stationary parts, and must have some clearance from it, inevitably a small fraction of the working fluid leaks through these small clearance passages from the high-pressure to the low-pressure region. In order to keep this leakage small the passages are made as narrow as possible and often tortuous and complicated as well; they are then called labyrinth seals. If the rotor vibrates or whirls relative to the housing, these passages vary in width and because they are so narrow in the first place, even a very small vibration may cause considerable percentage variation in width with consequent large percentage variation in leak velocity and pressure. To follow the details of all this by theory and to come out with the right phase angle is easier when the answer is known in advance than when it is not. In practice when such a case occurs and is properly diagnosed we know that the phase angle is wrong and that changes in the geometry of the labyrinth seal will affect the phase, so that a practical cure of the trouble is not particularly difficult. On the other hand we are not at present able to predict by theory whether a given design will be stable or not. Cases of this type are not common in turbines; in 30 years I have come across only three: a whirling steam turbine, a whirling vertical-shaft water turbine, and a longitudinally vibrating horizontal steam turbine where the rotor was moving along its own center line against the elasticity of the thrust-bearing support structure. All three cases had natural frequencies of the order of 15-50 c/s.

Recently another case was observed in a Ljungstrom type heat exchanger, which is a device for transferring heat from the stack gas to the atmospheric air prior to its entry into the combustion chamber of the boiler in a large steam-electric generating plant. The unit contains a rotor of some 30 ft. diameter and 15 ft. height, rotating about a vertical axis at approximately 1 rpm. The stack gas and the fresh air flow vertically through nests of vertical tubes in this rotor and the pressure differences are measured in inches of water. The rotating part is enclosed in a non-rotating housing so that some form of seal between the two is necessary to limit bypass leakage of the stack gas as well as of the air through the annular space between the rotor and the housing. The rotor, suspended from a thrust bearing at the top was found to sway sideways at its natural frequency of about 1 c/s. The consequent leakage passage width variations caused a pressure variation in the annular space (180° stack gas, 180° fresh air, separated by a baffle) of the order of a fraction of an inch of water only, but this very small pressure difference acting on an area of 30×15 ft.^{2} amounted to a considerable force. Again here there are many hundreds of these exchangers in operation, all apparently identical, and in only a few of them did this trouble appear, all others being satisfactorily stable.

The valves of Diesel-fuel injection systems have exhibited this type of vibration. When injection begins, the fuel pressure pushes the valve off its seat and while the fuel charge is rushing by, the valve

floats on its seating spring. If it vibrates (at high natural frequency, because the seating spring is stiff) it varies the width of the fuel passage and, as a consequence, it may become unstable. Since no reports have been heard of trouble with these valves for some years, it is assumed that the instability has been successfully eliminated.

Just after the 1939-45 war there was a short period of great enthusiasm and active development in the field of "ultrasonics," a name coined for the region of frequencies in excess of 10,000 c/s, by means of which it was proposed to homogenize milk, kill bacteria, produce emulsions, and do many other things. For this it was necessary to make ultrasonic vibration generators, which could put out high energies, and at least one such generator was built on the principle just described.

Flutter

Flutter [23] is the self-excited vibration of a structure (usually, but not necessarily, a part of an airplane) in a fluid flow where the energy source is the flowing fluid (usually air). The first cases that were observed, diagnosed, and explained occurred early in the development of the airplane, in the wood-canvas-and-wire days. They were named "aileron flutter" and consisted of an up and down bending vibration of the wing associated with a periodic rotation of the aileron about its pivot axis. Since the aileron's center of gravity was not located in the pivot axis, a vertical vibration of that axis caused an angular flapping of the aileron, which in turn led to a periodic change in the effective angle of attack of the wing with a consequent lift variation. If the phase between these variables was such that a maximum of lift occurred at the instant of maximum upward vibrational velocity of the wing, the air stream pumped work into the motion and made it unstable or self-excited. After some years of trial and error an effective remedy for this trouble was found, in the early 1920's, by "balancing" the aileron. This consisted of shifting some of its area and mass ahead of the pivot axis, thus placing the center of gravity in that axis, or even better by making the pivot axis a "principal axis of inertia" of the aileron. It was assumed that this disposed of the problem, but a few years ago it reappeared with transonic airplanes, where it is referred to as "burr"; there it is associated with shock waves, and it is altogether a much more complicated phenomenon than the old-fashioned low-speed aileron flutter.

Another type of flutter which was also observed quite early consisted of a combined vertical bending and torsional motion of the entire wing in which the aileron did not take part. This was and is called "classical wing flutter" (as opposed to "stall flutter" to be discussed in the next section), and its explanation did not appear until 1935 by Theodorsen.[24] Whereas the old aileron flutter usually occurred at such a low frequency that the aerodynamic forces on the wing were determined by the instantaneous value of the angle of attack of the wing and aileron (yielding to a theory of a "succession of steady states"), this was no longer true for classical wing flutter. Theodorsen derived expressions for the value of the sinusoidally varying lift force and moment on the wing caused by a sinusoidal variation of the angle of attack and of the vertical position of the wing. These expressions were complicated, containing the wind velocity V and the frequency ω in an intricate non-linear way. Assuming an undamped harmonic motion of the wing under the influence of this lift force and moment led to a pair of complicated algebraic equations in the two unknowns V and ω. The solution for these gave the flutter speed and the flutter frequency. The speed was such that below it a wing vibration would decrease in time, being damped; above it the vibration would grow, and at the flutter speed, by definition, the vibration would remain at constant amplitude, undamped.

The frequency of the flutter motion was found to be quite different from the natural frequency of the wing in still air, which places the flutter problem in a class by itself among self-excited vibrations. This is because the natural frequency is determined by a balance between spring and inertia forces and in most practical self-excited cases the exciting force is small with respect to the spring and inertia forces. In a fluttering wing, however, the exciting alternating wind force is of the same order of magnitude as the spring or inertia force and hence affects the frequency.

With this theory the critical flutter speed could be roughly calculated for each wing, and the result depended on several parameters, such as the size, the mass, and rotatory inertia, the bending stiffness and the torsional stiffness. After some experience it was found that among these variables the torsional stiffness was by far the most important one, so that in general the flutter speed could be raised by increasing the torsional stiffness of the wing structure. By this time canvas and wood had given way to aluminum wing structures of considerable thickness and the flutter situation was under adequate control in the period just preceding the 1939-45 war.

Flutter was not limited to the wings alone; it appeared as well in the rudder, elevators and tail surfaces and also in the propeller blades: in short, wherever some aerodynamic surface felt a change in its wind force as a result of its own vibratory motion. Once the phenomenon was well understood for a wing, flutter in other configurations could be explained when it appeared and eliminated by stiffening or other means. Then during the war and in the period thereafter the development of the airplane became explosive: in entering the subsonic and supersonic domains it became necessary for aerodynamic reasons to reduce the thickness of the wing sections, thus drastically decreasing the tor-

sional stiffness. Also the expressions for the harmonic lift force and moment caused by the vibration are entirely different from those for the subsonic case and what is more serious these expressions are not yet known theoretically. Thus before a flutter calculation can be attempted the forces as a function of the frequency have first to be determined by wind tunnel tests. Whereas in the simpler prewar airplanes an idealization of the wing into a "beam" was reasonably adequate, in the new types this is no longer justified and the frequencies have to be calculated on the basis of an idealization in terms of a "plate." These calculations (for which electronic computers are becoming more and more indispensable) are so intricate that the flutter problem has grown into a recognized science by itself with a name of its own "Aeroelasticity," with an extensive body of literature,[25, 26] and a large and growing group of specialized engineers.

Whereas the usual method of determining the flutter speed of a proposed design is by calculation on the basis of aerodynamic coefficients found by wind tunnel testing, it is also possible to do it entirely by model test. If a reduced scale model of the airplane, or of a representative portion of it, be built of the same materials and placed in a wind stream then the flutter speed of the model is the same as that of the full-scale airplane and the flutter frequency of the model is faster than that of the airplane in the size ratio. The dimensionless combination fc/V (where f is the frequency, c the "chord" or width of the wing, and V the flutter speed) preserves its value unchanged, and it can be recognized that it is equal to the number of full vibration cycles during the time it takes the wind to progress one wing chord c. This quantity (sometimes appearing in the literature with a factor π) is known as the "dimensionless frequency." The recent trend towards thinner and torsionally more flexible wings is expressed by the fact that the dimensionless torsional frequency $f_t c/V$ for fighter airplanes has been reduced by a factor of between 3 and 5 during the last 15 years.[26] Model testing for flutter is not generally practiced on account of the very real difficulties in producing true geometric reduction in size. If such a model is to be made at a scale of 1/20, it is theoretically necessary to reduce the skin to a thickness of 1/20 and to attach that skin to the spars with rivets of 1/20 size, and when the practical difficulties force us to a simpler "equivalent" construction, we can never be quite sure whether or not the modifications have affected the stiffness, and hence the flutter speed.

So far all authenticated cases of classical flutter have been in airplanes or missiles. In 1940 the Tacoma suspension bridge failed by wind-induced vibrations and for several years thereafter one of the possible explanations for the failure was in terms of flutter, which, however, was disproved by special wind-tunnel experiments[27] in the period 1945-50.

Another phenomenon that has been with us for some 50 years is the "singing" of ship propellers or of the rotors of water turbines, which is a vibration of a frequency of the order of 500 c/s of small amplitude, manifesting itself by a loud penetrating noise, but seldom leading to structural failures. Obviously it is some sort of self-excited motion in one of the higher modes of the propeller blades caused by the water stream. Many papers have been written on it with various explanations,[28] but no general agreement exists among the workers in the field on the exact mechanism of the phenomenon. A satisfactory practical cure has been found in sharpening the leading as well as the trailing edge of the blade.

Recently a case appeared of a destroyer, vibrating at its natural frequency of some 300 c/min, as a beam in a horizontal plane (so-called lateral vibration) in a three-noded mode. It did this only at high speed through the water, the vibration disappearing below 80 percent of the top speed. An explanation in terms of classical flutter of the entire hull was put forward but proven wrong by the discovery that the instability could be made to disappear by changing the length of the coupling rod between the two rudders. These rudders moved in unison, and the change in the rod length increased the relative angle between the two rudders by a few degrees. No precise detailed explanation is available; it may have been closely related to a classical aileron flutter, it may have been caused by a stalled condition of the flow near the rudders, or possibly by some other mechanism not diagnosed as yet.

Stall Phenomena

An aerofoil section, be it an airplane wing or a turbine blade, operating in a normal fashion with a sufficiently small angle of attack, experiences a lift force nearly proportional to the angle of attack. Imagine such a section mounted cantileverwise and vibrating at its lowest natural frequency in the fluid stream by bending in its limber plane. During the down beat of its motion it feels a small wind coming from below, which, compounded with the main wind stream causes an apparent angle of attack somewhat greater than the no-vibration angle and, consequently, the section experiences a lift greater than normal. Similarly during the upbeat of the vibration the lift is smaller than normal. Thus the excess part of the lift is always a force opposite in direction to the vibrating velocity of the section, which means that the motion is damped and the system is said to be dynamically stable. However, when the average no-vibration angle of attack surpasses some critical value of about 10° the flow breaks loose from the aerofoil which is then said to be stalled and in that condition the slope of the lift *versus* angle curve is the reverse from the previous case; that is, for an increased angle of attack the lift decreases. With this change in sign in the foregoing argument its conclusion is reversed, so that a vi-

brating aerofoil section in the stalled condition is dynamically unstable: the excess lift force caused by the vibration is always in the same direction as the velocity. Hence a stalled blade will vibrate by itself and this phenomenon is called "stall flutter." In airplane wings it is not of particular importance, because when stall occurs, the craft loses its lift as well as its control so that the pilot has more serious worries than just a little flutter.

In airplane propellers with fixed blades during take off (low speed of the airplane with fast rotation of the propeller) the angle of attack is large and certain parts of the blade become stalled. However, the damping provided simultaneously by the normally operating parts of the blade keeps the vibration from developing and the problem was never of great practical significance.

However, with the advent of jet-aircraft propulsion the stall problem in the blades of the axial compressor has become very serious indeed, and it is still very much with us at present. The starting point in the design of a compressor is the ratio between the rotor peripheral speed and the axial entrance speed of the air, and on the basis of this ratio all blades in the various stages can be designed with proper angles of attack for perfect operation. But during the take off, and in certain other essential operating conditions, the ratio between these two speeds is necessarily different and hence the angles of attack are modified. Thus the design must be made as a compromise between the various operating conditions, and we are forced to accept situations where some rows of blades operate close to the stall angle while others do little work having a very small angle of attack. Such a compromise design can easily be carried out satisfactorily for compressors of moderately small compression ratios of, say, 3/1. For increasing compression ratios it becomes more difficult and finally impossible.

At least two methods of designing around this difficulty have been devised and have been incorporated in current engines. The first and most drastic of these is to cut the axial compressor in two parts, running it on two shafts (one passing through the bore of the other), each of which finds its own appropriate rpm, depending on the operating condition. This is called a "double spool engine" and with it the problem of a compressor of a ratio 9/1, which it would be impossible to design properly on one shaft, is reduced to that of two compressors of ratios 3/1 each. The second method is to modify the angle of the stream entering each row of rotating blades by making the non-rotating guide vanes angularly adjustable.

When in operation a row of blades actually reaches the critical stall angle, violent stall flutter is inevitable, and the blades will surely fail by bending fatigue, which makes the avoidance of this condition a complete necessity. But when the overall angle of attack on a row of blades is somewhat

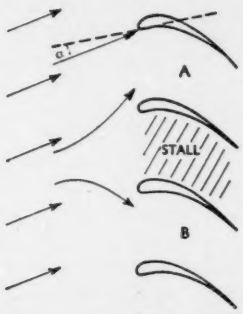


Figure 2. Explanation of the Mechanism of Rotating Stall in the Compressor of an Aviation Jet Engine.

smaller than this "critical angle for fully developed stall," something less serious happens which is known as "rotating stall." The passages between two, three, or four adjacent blades go into a stalled condition and this block of stalled passages moves relative to the stator in the direction of the rotation with roughly half the speed of that rotation (and hence relative to the rotor it moves at half speed in a direction opposite to that of the rotation). Sometimes one such block exists, sometimes two or even three, all moving separately with respect to the rotor at the same speed; the passages between these blocks having smooth flow without stall. A tentative sketchy explanation (Figure 2) is that the flow resistance in a stalled passage is greater than in a smooth one, the stalled air acting like a partial plug. Then the approaching air tends to flow around the obstacle, which in the passage A tends to increase the angle of attack and in B tends to decrease it. Thus the passage A will stall next and B tends to become smooth if it were stalled previously. Hence the "plug" will move in the direction opposite to that of the rotation relative to the rotor, but so far the explanations of why that should happen at roughly half the speed of rotation have not been entirely satisfactory.[29] Research on the subject is going on in various places at this time.

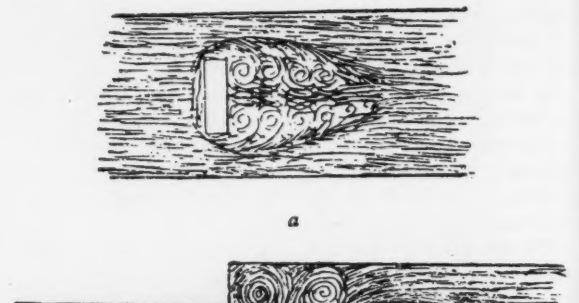
An individual blade in a case of rotating stall will be alternately stalled and smooth in a reasonably rapid succession. Such instability as the stall itself might cause will not persist long enough in time to build the vibration up to a large amplitude, and during the subsequent period of smooth flow the motion will be damped out again. Thus the probability of structural failure from the stall instability itself is not great. However, when an individual blade is suddenly enveloped by the stall it experiences an impulse, and since the plug or plugs of stalled air move relative to the blades at almost constant speed, each blade is subjected to periodically recurring impulses, which if their frequency is close to a natural one may cause failure by resonance. Since the number of plugs and their relative angular location cannot as yet be predicted, the

condition of rotating stall should be avoided if at all possible. This imposes a severe handicap on the designer, so that at the present time the phenomenon of rotating stall sets a limit to the compression ratio that can be built into a single-shaft axial compressor.

For the past 30 or more years large high-voltage electric-transmission lines under certain climatic and wind conditions have been plagued sporadically with a vibration known as "galloping," which has been partially explained as a stall flutter.[30] It appears once a year or so in Canada and in the northern United States, when the lines are sleet-coated and a wind of at least 15 miles per hour is blowing perpendicularly to the line. The cross-section of a sleet-coated wire may take various forms, none resembling true aerofoils but some of these shapes will show a greater lift or vertical force from the wind when the angle of attack is diminished. Thus a stall instability may occur. The result is a vertical vibration of the entire span at a slow frequency of about 1 c/s and a large amplitude of up to 10 ft. at midspan. In all these years no simple practical cure has been found, except the questionable one of melting off the sleet by partially short circuiting the lines. The explanation in terms of a stall flutter associated with purely up and down motions is probably too simple, because torsional motions of the wires have often been observed to accompany the more spectacular vertical ones. The National Research Council of Canada in co-operation with the electric power companies of Ontario and of some of the adjacent American states continue research on this subject.

Kármán Vortices

When a cylinder, not necessarily of circular cross-section, is placed in a stream flowing perpendicularly to its axis, vortices are formed which break away from the cylinder alternately from the two sides, causing an oscillating force in a direction perpendicular to the stream velocity and also perpendicular to the cylinder axis. There are good drawings of these eddies (Figure 3) in Leonardo da Vinci's notebooks,[32] but the first description in modern terms came four centuries later by Strouhal (Rayleigh's "Theory of Sound," vol. 2, Section 372), who expressed the frequency of eddy shedding by the formula $f=0.22V/D$, which means that one full cycle of the sidewise force variation occurs during the time that the flow progresses $1/0.22=4.5$ diameters of the cylinder. The dimensionless quantity FD/V is now known as the "Strouhal number"; it is about 0.22 for circular cylinders and assumes values not far removed from 0.22 for other cross-sections. In 1911 von Kármán [32] published a paper on these vortices explaining them in so much more detail than had been done previously that they have been named after him ever since. During the past 30 years this phenomenon has given much trouble on various structures and has been responsible for



a

b

Figure 3. Two Sketches by Leonardo da Vinci on the Vortex Wake Behind a Rectangular Obstacle.

the total destruction of a major engineering work—the Tacoma bridge. In ordinary circumstances the alternating force from the eddies is of no particular interest: only when the Strouhal frequency happens to coincide with a natural frequency of the structure on which the stream acts, and resonance takes place, does the phenomenon assume engineering importance. This has occurred on electric transmission lines, submarine periscopes, smoke stacks, a roofless oil storage tank and . . . the Tacoma bridge.

In a transmission line of $D=2$ in. diameter and a wind speed of 20 miles per hour or 30 ft. per sec., the Strouhal frequency comes to approximately 40 c/s, which corresponds to a high mode of natural vibration of the line showing a configuration of many (say, 20) sine waves along a single span. In this mode the line vibrates at an amplitude of a fraction of an inch resulting in frequent bending fatigue failures. A good practical remedy exists in the form of inertial friction dampers which are mounted on the line at some 6 ft. distance from the point of attachment to the towers. It will be appreciated that this motion of the transmission line is fundamenally different from the "galloping" motion caused by stall flutter, discussed previously.

During the 1939-45 war annoyance was experienced by submarine captains, because during a sneak-approach to a target at a speed of some 5 knots, the eddies in the water acting on the extended periscope tube (of about 8 in. diameter) with a frequency of 2 or 3 c/s came to resonance with the natural cantilever frequency of the periscope and the resulting sidewise vibrations blurred the target picture. One method of eliminating the trouble would be to increase the tube diameter, which by bringing the Strouhal frequency down and the nat-

ural frequency up would push up the critical speed out of the operating range.

Smoke stacks will sway sidewise in a steady wind of the proper critical speed if their internal damping is sufficiently small. Brick or concrete chimneys have never exhibited this vibration as far as we know, but welded steel stacks are all prone to it. In the very large ones the situation has now reached the danger point; a few years ago a welded stack of 16 ft. diameter and 300 ft. height, vibrating in a 50-knot wind at a frequency of about 1 c/s, actually failed by local buckling and cracking. Stacks of this size are not safe without some provision for increased damping, either by lining them with concrete inside or by the use of friction dampers in guy wires.

The expression $0.22V/D$ for the Strouhal frequency applies to circular cylinders that do not vibrate. When a chimney vibrates by resonance, its elastic and inertia forces are many times greater than the Kármán wind force, so that the frequency at which it can vibrate is its natural frequency only. It had been suspected for many years that if a cylinder vibrated at a frequency somewhat different from the natural Strouhal frequency of the eddies, the eddies could be forced by the cylinder motion to take place at the cylinder frequency, and thus still excite the cylinder even at a frequency different from Strouhal's. This vague belief was cleared up recently by experiments of C. Scruton at the National Physical Laboratory,[33] which showed that for very small damping a chimney could be excited at frequencies up to twice the Strouhal frequency. This means that a stack may be in trouble in a range of wind speeds extending between the calculated Strouhal critical speed and twice that value. The results of that investigation are shown in Figure 4.

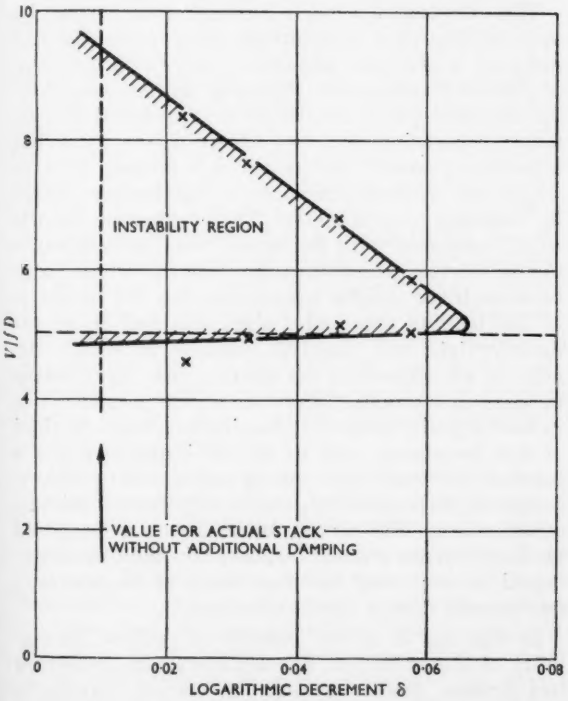


Figure 4. Stability Diagram of a Circular Steel Smoke Stack in the Wind, Obtained by a Model Test in the National Physical Laboratory.

Reproduced from Scruton.[33] The ordinate V/fD is the reciprocal of the Strouhal number and the abscissa is the "logarithmic decrement" or percentage decay in amplitude per cycle of the stack in still air.

The most spectacular case of a vibration caused by Kármán vortices occurred in 1940 on a suspension bridge in Tacoma, Washington, in the Pacific Northwest of the United States. The cross-section of the bridge was a horizontal I of 39 ft. width with two 8-ft.-high vertical flanges of solid steel. Under a side wind of 42 miles per hour, the bridge became unstable torsionally, vibrating with angles up to $\pm 45°$ at the quarter and three-quarter span points with torsional nodes at midspan and at the two ends, resulting in a complete structural failure. For several years after the event, the exact mechanism of the cause remained unknown until by a series of wind-tunnel model experiments it was shown to be due to Kármán vortices coming off the sharp top and bottom edges of the vertical flange facing the wind.[27] In the newly reconstructed bridge three changes were made:

(1) The solid vertical flanges were replaced by open trusses through which the wind could blow freely;

(2) Longitudinal slits were provided in the horizontal bridge deck between traffic lanes, preventing the possibility of pressure differences between top and bottom at those points;

(3) The "open" I-section was replaced by a "closed" rectangular box section (with three of the four sides being open trusses) which torsionally is a hundred times as stiff, thus raising the torsional natural frequency well above the Kármán frequency at any conceivable wind speed.

This last remedy was subsequently applied to the San Francisco Golden Gate Bridge and to some other bridges which had shown very mild tendencies towards wind vibration. In this connection it may be of interest to remember that about a century ago Stephenson proposed a suspension bridge with a box girder, and that in the final design Fairbairn preserved only the box and eliminated the suspension feature.[34] In the course of the next 100 years, bridges were gradually made longer, lighter, and more flexible, until suddenly the new limitation of the Kármán vortices was met, which reintroduced the box structure and put a stop to the trend towards greater flexibility.

Shimmy

The Oxford dictionary asserts that "shimmy" is an American term, born in 1919, signifying a "kind of fox-trot accompanied by tremulous motions of the body." In a past static age our profession adopted into its technical vocabulary many words relating to the body such as footings, knees, arms, ears, eyes and teeth, and it is fitting that in this new dynamic era the list should be augmented by an obviously dynamic word. The next edition of the dictionary might add another definition such as: "techn. (1925); an angular vibrating motion about a vertical axis of a wheel in a moving vehicle resulting in a sinusoidal path of the wheel on the ground."

This type of motion was first observed in the castored small wheels of hand push carts in factories, and did not receive any particular attention. Then, in the late 1920's, it appeared in the front wheels of automobiles, brought about at that time by a lowering of the air pressures in the new balloon tires. The two front wheels were rotating on shaft stubs integral with a common front axle, on which the car body was elastically supported. If on an uneven road one wheel would go up, passing over an obstacle, the front axle would tilt and the two front wheels would rotate through a small angle about a fore-and-aft axis along the center line of the car (a "rolling" motion). Since at running speed the wheels act as gyroscopic bodies, such an angular vibration about a fore-and-aft axis is gyroscopically coupled to a vibration about a vertical axis, which is a shimmy motion. This was a nuisance for a number of years, and it was attacked by the introduction of dampers retaining the basic geometrical construction, until a fundamental cure was found in mounting the two front wheels independently, each on a parallelogram linkage so that the up and down motion of the wheel could take place in its own plane without rolling rotation. This destroyed the coupling between the up and down motion and the angular shimmy motion and constituted a complete remedy of the trouble. At the time, during the early thirties, an extensive literature appeared on the subject, particularly in the automotive journals, which is now of historic interest only.

During the 1939-45 war, trouble was experienced in the castored tail wheels of aircraft landing gears by the Germans and the Allies alike, both sides of the conflict working hard on it and producing many reports which were kept secret from the enemy and consequently equally secret from almost everybody on the friendly side as well. The trouble could be eliminated on an empirical basis by torsional dampers or by changes in dimensions, and these practical cures were often accompanied by official reports covering the theory of highly simplified systems, which later proved to be non-representative of the truth. More recently, with the explosive increase in weights, sizes, and landing speeds of airplanes, the trouble has reappeared in a very serious manner, this time in the swivelling nose wheel.

An extensive theoretical and experimental analysis of this phenomenon by W. J. Moreland [35] showed that simplifications in terms of two or three degrees of freedom failed to stand the test of experimental verification. In order to do justice to the problem, it was necessary to consider four degrees of freedom in the landing gear alone, apart from the airplane of which it forms a part. These were: (1) the shimmy angle of the wheel; (2) the lateral or sidewise deflection of the vertical strut due to its elasticity; (3) the sidewise elastic deflection of the rubber tire; and (4) the torsional angle of the dashpot or damper structure, which necessarily has to be made very light.

The result of the analysis has to be expressed in terms of a number of significant parameters, such as the landing speed, the castor distance or "trail," the radius of the wheel, the mass and moment of inertia of the wheel as well as of the strut and other parts, the constants of the dashpot and of the rubber tire, the various structural stiffnesses and the weight load on the wheel. It was found that 15 such parameters were needed to describe the important properties of various landing gears.

The calculation of such a complicated system by ordinary means is prohibitively time consuming and becomes a practical possibility only with the help of electronic computers. Moreland and his associates during some years calculated several hundred systems on the computers. In addition they constructed a versatile, small-scale model of a landing gear in which the various constants or parameters could be changed conveniently. The computed results when compared with the model tests showed satisfactory correlation. Even this is not the whole story because these results would describe the behavior of the landing gear only when attached to an infinitely rigid and massive airplane, of which the point of attachment of the landing gear strut moves forward in a straight line, even in the presence of a violent shimmy at 30 c/s. No airplane frame is rigid at this frequency, and its lateral impedence (as a function of frequency) can be calculated for a new design or determined experimentally for an existing construction. The actual behavior of a proposed landing gear on a given airplane can then be determined by matching the impedance of the airplane frame with that of the landing gear.

In this way it is now possible to predict the stability of a new design. The old empirical procedure had broken down, because for certain conditions roughly described as wheels of large diameter and inertia, it was proved impossible to eliminate the instability by torsional damping alone, so that no matter what was done in the way of damping, shimmy would occur. Cases of this type can be made to behave satisfactorily only by a proper design of sev-

eral other parameters such as size, mass, and stiffness.

A phenomenon closely related to shimmy is the "nosing" of locomotives or other railway carriages on their tracks, in which the vehicle sways sidewise, sometimes quite violently, at a frequency of about 1 c/s or slower.[36] This problem was of great practical importance in the period from 1910 to 1935, but now it seems to have been solved satisfactorily by proper design.

Instability of Combustion in Rocket Engines

A rocket is a device consisting of a combustion chamber and a nozzle through which the combustion products are expelled rearward at high velocity. They are being made with solid or liquid propellants; in the solid-propellant type the fuel and oxidizer are premixed, as in gunpowder, into a solid block, which is made to burn on one face; in the liquid-propellant type the fuel (for example, gasoline) and the oxidizer (for example, liquid oxygen) are carried in separate tanks and are pumped to the combustion chamber where they are mixed and burned together. The solid-propellant rocket is very ancient and has been popular for a thousand years in Chinese New Year celebrations; the liquid-propellant type is newer, having been introduced to modern civilization under the sinister name V2.

It has been known for many centuries that Chinese toy rockets can be made to scream loudly at a high pitch, and this effect under the name of "pyrotechnic whistle" has been used in war, sometimes psychologically to frighten the enemy, and also as a decoy for acoustically homing underwater torpedoes. In a screaming rocket or pyrotechnic whistle the combustion is not smooth but has fast alternations in intensity which are accompanied by fluctuations in pressure in the combustion chamber. If the rate of combustion is a function of ambient pressure, a fluctuating pressure causes a fluctuating combustion and consequent gas production, and hence a fluctuating pressure again. This is a typical feedback mechanism, which becomes unstable if certain phase relations occur.

About a century ago illuminating gas was one of the new wonders of engineering; it was used in open jets, of which the combustion under certain circumstances was found to be unstable. These "singing flames" were made quite fashionable in London society by marvellously spectacular demonstrations and lectures in the Royal Institution by its Professor John Tyndall.[37] Some years later Rayleigh wrote his great book and commented as follows ("Theory of Sound," vol. 2, Section 322g): "If heat be periodically communicated to and abstracted from, a mass of air vibrating (for example) in a cylinder bounded by a piston, the effect produced will depend upon the phase of the vibration at which the transfer of heat takes place. If heat be given to the air at the moment of greatest condensation, or be taken from it at the moment of greatest rarefaction, the vibration is encouraged." This was meant by Rayleigh to apply to Tyndall's singing flames. Not many years thereafter, however, the heat so communicated and so abstracted in a "cylinder with a piston" was indeed "encouraging the vibration" of the pistons of Diesel engines to propel ships across the oceans of the world.

With the recent development of rocket engines, particularly with those of very large size and thrust, this instability has come to us as a very serious technical problem.[38] The phenomenon starts suddenly without warning; the pressure fluctuations in the combustion chamber may be as high as ± 100 percent of the mean pressure, the frequency as high as several thousand cycles per second and often it is all over in a fraction of a second with the machine in pieces. It is not healthy to be near a large rocket when it is operating, so that all observations have to be made by instrument at a safe distance, and obviously it is not easy to contrive high-frequency sensitive pressure pick-ups in the interior of a flame, so that experimental research into this problem is fraught with extraordinary difficulties.

The first appearance of this trouble in the United States occurred, immediately after the termination of the war in 1945, on a liquid-propellant rocket engine. It was of a relatively low frequency, about 100 c/s, of relatively low intensity, the pressure variation being about ± 25 percent of the mean, and hence not destructive. Adequate measurements could be made, showing that the pressure variations extended into the feed lines of the fuel as well as of the oxidizer. It was reasoned that at the moment of high pressure in the combustion chamber the inflow of fuel and oxidizer was impeded. If then at this instant of high pressure there would be less fuel supply and less combustion and consequently less heat generation, the phase angle of Rayleigh's rule would be just wrong, and the system would be completely stable. However, the combustion itself takes time, and the flow of the liquids through the pipe system also takes some time, so that the instant of minimum heat generation lags behind the instant of maximum pressure by some (short) time. Rayleigh now states that if this time lag lies between 1/4 and 3/4 period of the fluctuation (or between 5/4 and 7/4; 9/4 and 11/4, etc.) the phenomenon is unstable. This is a reasonable explanation of the observed instability and for sufficiently slow frequencies the time lag can be calculated after a fashion. At very slow frequencies of the order of 1-10 c/s this effect has been observed in boilers or industrial oil-fired furnaces when the air supply to the burner is choked and a good practical remedy then is to admit some extra air into the furnace, bypassing the burner blower. Similarly the observed instability of rocket engines (in the "low frequency" range of 40-200 c/s) can be remedied in practice by making changes in the supply lines,

pumps or nozzles and hence, although the real fine points of the theory are not completely understood, the situation is acceptable.

However, with the high-frequency type of rocket-engine instability (from 800 c/s up), there is at present no such practical cure available. Experiments have shown that in these instances the pressure fluctuations are practically limited to the combustion chamber itself and hardly penetrate into the feed system. Here then we have a combustible mixture of gas (which is kept permanently combustible by the inflow of fresh gas and the outflow of burnt gas) which can vibrate in any one of the many possible natural modes of vibration of a gas in some sort of chamber. If the rate of combustion of the gas would reach a maximum at the same instant as the pressure (as seems natural to assume), then we have satisfied Rayleigh's rule or in other words we have the situation in a Diesel cylinder or in a gas-turbine cycle and, consequently, work is done on the vibratory cycle which will then continue for the same reason that a Diesel engine or a gas turbine keeps on running. If this were the case, then all possible modes of motion of the gas in the combustion chamber would be unstable and no rocket engine could run smoothly. Evidently this is not the case and there must be a time lag between the maximum pressure and the maximum combustion. For stability this time lag must be at least 1/4 period, or $1/8\times10^{-3}$ s in a 2000 c/s vibration. The detailed precalculation of this is not possible at this time on account of the many unknowns in the combustion process. Vigorous researches on the fundamentals of combustion are now being pursued in most countries, and in this connection it may be of interest to quote another sentence from Rayleigh (vol. 2, Section 322h): "Partly in consequence of the peculiar and ill understood behavior of flames, the thorough explanation of the phenomena now under consideration is a matter of some difficulty."

VARIOUS OTHER CASES

In conclusion a number of cases of a miscellaneous nature that have been of recent interest will be briefly mentioned.

The field of ultrasonics [39] consists of the applications of frequencies above the audible range (above 10,000 c/s), which are usually and most conveniently generated by electronic techniques. Its development started with underwater signalling devices during the 1914-18 war, remained more or less stagnant during the twenties and thirties and received a vigorous boost again during the 1939-45 war with the creation of "Sonar" and similar techniques. An unexpected by-product of this work was the explanation of the old question of the flying of bats in the dark as an ultrasonic sonar-echo mechanism. The experience gained during the war has led to some peacetime applications, principally in measuring techniques.

Ultrasonic waves are now being used for detecting cracks or flaws in metal structures, particularly in welded joints, by moving a noise generator over the surface from point to point.[39] When this instrument passes over a sudden change in thickness of the steel plate or structure or when it passes over an interior crack or non-metallic inclusion, there is an indication on a meter. The use of this method of inspection is increasing, and it is becoming a serious competitor to the older and more familiar X-ray technique.

The measurement of the violent and rapid temperature changes in a cylinder of an internal-combustion engine during the small fraction of a section immediately after the spark ignites the gas is obviously a difficult problem. It was accomplished recently [40] by sending an ultrasonic wave across the cylinder and measuring the time taken between the start of the signal and the return of the echo from the opposite cylinder wall, thus determining the velocity of sound in the gas, and by direct correlation, its temperature.

In systems involving a servomechanism small changes in the value of the quantity to be controlled or kept constant result in operations by the servomotor with considerable power and this reacts directly on the variable to be controlled. Thus if certain unfortunate phase relations occur, the system may become unstable. Consider, for example, the automatic steering mechanism of a ship or airplane. A gyroscopic direction sensor is mounted in the vehicle and when this sensor finds a small deviation between the actual direction of the ship and the desired direction, it causes power to be applied to the servomotor controlling the rudder. Now imagine the vehicle to be vibrating in one of its natural bending modes and let the gyro-sensor be mounted in such a location that it experiences a vibrating angular motion. Then the rudder will receive vibrating impulses at the same frequency and the phenomenon may become unstable. In principle this may occur at any one of the natural modes of the system, even the higher ones with fast frequencies. In an ocean ship this possibility is quite excluded because the steering engine, be it steam or electrically operated, is far too slow and under the influence of a rapid alternating signal the rudder would simply stand still in the neutral position. Also the angular amplitude of motion to be expected in a ship at the location of the gyro-sensor should be extremely small. In an automatic pilot of an airplane the possibility of such an instability is less remote and when the system is made more flexible yet and is required to respond to fast signals as in some guided missiles, the possible appearance of self-excited vibrations of this type must be considered seriously during the design stage.

Our most modern power plants actually operating involve nuclear fission reactors, and although

there are indications that this type of plant may become obsolete before it has properly grown up (in favor of some sort of fusion process), we will assume that nuclear fission reactors will remain of interest to mechanical engineers for some time to come. Several types of this reactor are being developed,[41] one of which is the boiling-water reactor. The water acts as a "moderator," which means that the fissionable material will react and generate heat only when it is surrounded by water, and in the absence of water the reaction dies out immediately. In a properly operating boiling-water reactor the radioactive material is surrounded by water with a uniform distribution of small steam bubbles. Suppose now that the reactor momentarily acts at greater than normal intensity. Then it will rapidly boil the surrounding water, and soon it will find itself in an atmosphere of steam, with the result that the reaction stops and no more heat is generated. The steam rises up and water rushes back in its place, thus reigniting the atomic fuel, and so on. Instabilities of this type have been observed and they are of an intensity awesome to a person accustomed to ordinary mechanical vibrations. The maximum rate of power generation (in kW) is 500 times greater than its minimum value and these alternations take place slightly faster than 1 c/s.[41, 42]

The last vibration problem to be mentioned here is a biological one. Helmholtz [43] explained the physical mechanism of the ear [44] and of the vocal chords, organs of surpassing beauty and ingenuity of which our wonder and admiration increase the more we study their details. But their operation belongs to the realm of acoustics, which was not included in the subject matter of this lecture. The human heart, however, is a vibrating organ functioning at a frequency which does fall within our domain. A few years ago a specialist surgeon was considering an operation on a heart in which the central valve was leaking as a result of a history of "rheumatic fever." It was proposed to close the valve permanently by surgery and to connect the two chambers of the heart by a bypass consisting of a short section of artery in which an artificial mechanical check valve was to be inserted. Among the many very difficult design problems of such a valve most were biological and some mechanical, and not the least one was that of the fatigue and wear strength of the element. While this was being discussed it was remarked that the human heart during a full lifetime performs three thousand million beats, a number far greater than that of any valve or other vibrating or reciprocating part in any machine made by human beings.

Thus, in the presence of the very great recent accomplishments of our engineering profession and of science in general, we stand ever more humble and admiring in the contemplation of Nature's craft.

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RUSSELL PYLES

CYLINDER ECONOMICS WITH TODAY'S FUELS, LUBRICANTS AND MATERIALS

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is Vice President and Chief Engineer of the Van der Horst Corporation of America, having been associated with the corporation since 1942. Educated at George Washington University and Temple University, he has been associated with engine development since graduation from Air Corps Flying School, Kelly Field, in 1922. From 1922 to 1929 he served with the National Advisory Committee for Aeronautics as an associate mechanical engineer, subsequently joining the Diesel-Electric Division of Westinghouse Corporation to assist in the development of locomotive engines and later associated with Chicago Pneumatic Tool Company and Clark Bros. Co. in the development of diesel and gas powered engines and compressors. He is a member of the American Society of Naval Engineers, the Society of Automotive Engineers, the American Society of Lubrication Engineers and the American Society of Mechanical Engineers, serving on the Internal Combustion Engine Code Committee of the latter. In addition, he served on the Committee on Engine Metals and Wear of the American Society for Metals. During his years of association with the design development and application of internal combustion engines he has authored a number of papers on engines and related subjects.

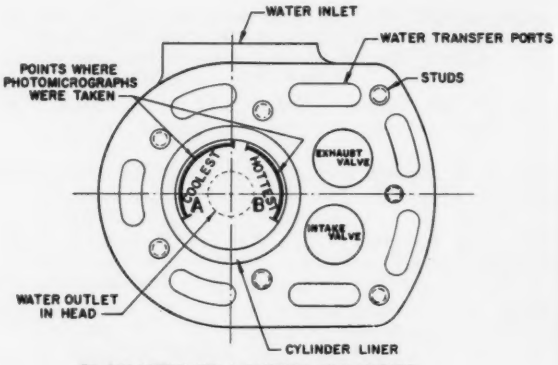
THE INCREASING use of heavy fuels and the recent development of corrosion inhibiting lubricants make it desirable to re-evaluate the economics of cylinder materials from a wear standpoint with special reference to chromium plated cylinders and cast iron in general. Concurrent with lubricating oil developments, there have been great improvements in plating technique which permit the economic deposition of chromium layers of two to three times the thickness which has heretofore been considered as the practical limit. The situation has been forced by the always present necessity of cutting cost, in this instance with cheaper fuels. It must be accepted that heavy fuels are more corrosive than lighter diesel fuels and that the accelerated wear encountered by their use is due mainly to corrosion. Neither high engine cooling water temperatures, choice of cylinder materials nor conventional lubricants, nor even the additive types up to Series III, have alone given a satisfactory solution to the problem. Each of these factors has been of some help and combinations have been a little more helpful—but not enough.

COOLANT TEMPERATURES

Raising cylinder wall temperatures above the dew point by higher discharge water temperatures has its limitations. Both cylinder wall temperatures and the dew point temperature vary throughout the stroke but experience shows that corrosion attack is first in the upper part of the stroke. There are ample reasons to believe that 70° to 80°C (158°-176°F) discharge water temperature is not high enough to eliminate condensation and corrosion in the upper cylinder when Bunker C or residual fuels are used. On the other hand, some experience with vapor phase cooling and pressure cooling with coolant temperatures of the order of 225°F indicates

beneficial results in reducing corrosive wear. However, the space requirements [1] for auxiliary equipment to accomplish vapor phase cooling preclude such an installation in an existing ship. It is much simpler to raise coolant temperatures by pressurizing the cooling system as this can be done with minor plumbing additions and alterations.

Objections to such increased temperatures may be raised by both engine builders and operators on the score of increased heat stresses in parts such as heads, pistons and cylinders as well as increased radiation to the engine room. The first objection is not valid because heat stresses are a function of temperature difference and not temperature alone. In general, the higher coolant temperature will tend to reduce temperature differences and lower thermal stresses [1] of critical engine parts. The greater heat radiation of engine parts to the engine room is more serious. With an increase of 50°F in the coolant temperature, water jacketed parts may be nearly this much hotter on their outside exposed surfaces and too hot to touch. Engine room temperatures will rise appreciably and require greater deliveries of cooling air to maintain tolerable working conditions. The full application of either of these expedients is impractical; however, the advantages of greater coolant temperatures can be partially obtained by temperatures of 71°-77°C (160°-170°F) which can be tolerated.



PLAN VIEW OF LAUSON CYLINDER

FIG. 1A

Figure 1A. Plan View of Lauson Cylinder.

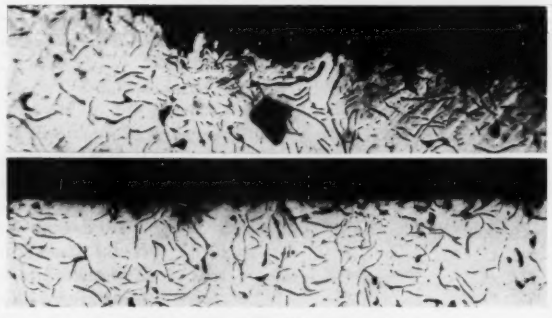


Figure 1B. Magnification 100 ×.

MATERIALS

The corrosion resistance to sulfuric acid of various cast irons commonly used for cylinders varies little, except for highly alloyed nickel and chromium compositions such as Ni-resist. Cost of such alloys becomes excessive and unfortunately, as corrosion resistance increases, the inherent normal bearing characteristics suffer. In spite of the enormous amount of development work done on various cast iron compositions for improved wear and corrosion resistance, this approach offers little hope. The presence of sulfuric acid promotes graphitization, an electrochemical reaction resulting in dissolution of the iron constituents with the exception of the graphite. The constituents are arranged from cathodic to anodic as follows[2]: ferrite, pearlite, pearlitic-phosphide eutectic, cementite and graphite. A good cylinder iron [3] is generally considered to have a pearlitic base, random graphite orientation, and a very minimum of ferrite. The maximum galvanic couple is set up between the ferrite-pearlite and the graphite. Corrosion occurs contiguous to each graphite particle, the corrosion product being black, soft and easily scraped by a knife or piston rings. The result is not readily seen on cylinder walls except by microscopic inspection but is evidenced by rapid dimensional change. Figure 1 shows this condition produced in a small engine cylinder by operating at 100° discharge water and gasoline doped with thiophene to the equivalent of 2.8 per cent sulfur.[4] Figure 2 is a 250 magnification of Figure 1. The dark grey graphite particles can be seen in the black matrix of corrosion products, and the path of corrosion follows the graphite flake. A similar electrochemical action will occur with nodular iron wherever a graphite nodule is exposed.

While chromium has much greater corrosion resistance to low concentrations of sulfuric acid than cast irons, it is nevertheless not immune and the effects can be seen in an engine cylinder with the naked eye. The corrosion in this case is chemical and not electrochemical because the exposed surface is a single homogeneous metal and there is no intercrystalline potential difference. When a chromium plated cylinder has been operating under corrosive conditions, it will show portions of the cylinder wall of a typical shiny chromium appearance and black patches of apparent lacquer. This apparent lacquer is generally water soluble and can be wiped off with a water wet cloth. Under this black corrosion product will be found a light grey, dusty, etched chromium, the surface of which is slightly below that of the adjacent brighter chromium on which the rings and pistons have been riding. Such a cylinder of 21½ inch bore is shown in Figure 3 after cleaning. The corroded patches are

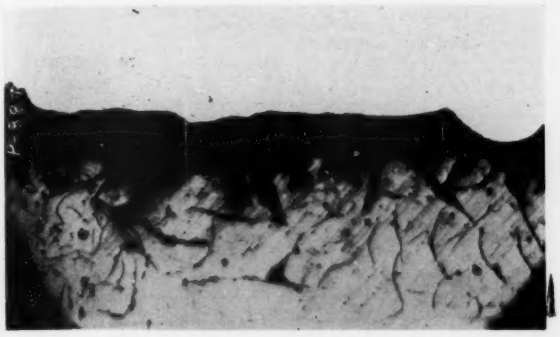


Figure 2. Magnification 250 ×.

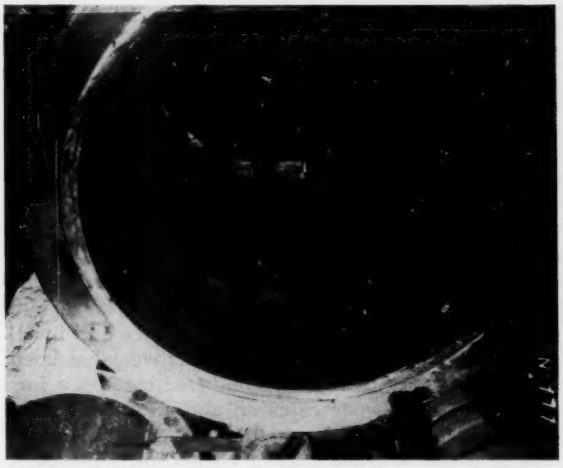


Figure 3.

light grey in the photograph. Contrary to cast iron, such corrosion is readily visually detectable in a chromium plated cylinder.

This easily recognized corrosion can also be a clue to the efficiency of cylinder lubrication and oil distribution. Figure 12 shows this condition in a cylinder after 4000 hours operation on diesel fuel with a heavy duty lubricant and the subsequent change in corroded areas after 2000 additional hours using a corrosion inhibiting type lubricant. Note the diminished size of the corroded patches, and that the rings are beginning to approach full circumferential bearing instead of riding on washboard ridges. While this cylinder has six oil holes with oil grooves, it is apparent that distribution is ineffective; corrosion did not occur in line with the oil holes but is a maximum between the oil holes. This engine does not have timed lubrication. In spite of poor oil distribution the corrosion inhibiting lubricant has effectively allayed corrosion, the cylinder is healing, and in another 2000 hours the piston and rings should have a full bearing and the wear rate will settle down to an attractive low figure. In measuring such corroded cylinders, it is necessary that measurements be made from peak to peak and valley to valley, i.e., in line with oil holes and half way circumferentially between them. If measured only at 90 degrees, it may appear that the cylinder is out of round which is not the whole story. The high and low spots are difficult to see on cast iron because it does not have the color contrast visible on chromium.

GENERAL WEAR RATES

Chromium plated cylinders have shown various results, favorable and occasionally unfavorable when operating on heavy fuels but interpretation is difficult and confusing because the wear results usually have shown the combination wear of the initial chromium and that of the cast iron after wearing through the chromium. Rational consideration requires that these respective wear rates be

TABLE 1[5]

Ship	Engine Type	Wear per 1000 hours Porus-Krome		Wear per 1000 hours Cast Iron		Total Hours	Total Wear		Cylinder diameter	
		mm	in.	mm	in.		mm	in.	mm	in.
8	2ssa			.65	.0256	6,000	4.0	.1574	740	29.133
17	2sda			.75	.0295	6,000	4.5	.1771	720	28.346
19	2sda			.70	.0276	8,000	5.5	.2615	780	30.708
A9	2ssa			.53	.0209	5,000	2.65	.1043	600	23.622
A10	2ssa			.55	.0217	9,000	5.0	.1968	600	23.622
11	2ssa	.25	.0098			4,000	1.0	.0394	740	29.133
A7	2ssa	.15	.0059			2,500	.37	.0146	680	26.771
A8	2ssa	.11	.0043			4,000	.45	.0177	600	23.622
A9	2ssa	.17	.0067			4,000	.7	.0275	600	23.622
A11	2ssa	.1	.0039			2,500	.25	.0098	670	26.377

WEAR—HEAVY FUEL—STANDARD LUBRICANTS

Average wear rate:
Porus-Krome .154 mm or .0060 in.
Cast Iron .63 mm or .0248 in.

Wear ratio:
Cast Iron : Porus-Krome = .63 mm : .154 mm
Cast Iron : Porus-Krome = .0248 in. : .0060 in. = 4.1 : 1

Porus-Krome cylinders have NOT worn through the chromium

separated and it can be done from available data. It must be conceded that the important point is the life of the cylinder no matter what material is wearing, but the optimum chromium thickness is tied up with both wear rates and the practical total permissible wear as determined by the engine itself and the factors of blowby, ring breakage and power output. Table 1 shows wear rates on a number of ships with cast iron and chromium plated cylinders but on none of the engines with chromium cylinders has the chromium worn through to the cast iron. The plating practice at that time was to deposit .8 mm (.0315 in.) of chromium on the diameter and in all cases shown in Table 1 the chromium wear has been less than .8 mm (.0315 in.). Thus we can get a fair idea of average wear rates on chromium only and cast iron. The data shows that the cast iron wear rate is about 4 times that of chromium only, both operating with heavy fuels and under corrosive conditions. This is confirmation of the relative corrosion resistance of chromium to such concentrations of sulfuric acid as occur in an engine cylinder. This also leads to the conclusion that much thicker chromium may be desirable but there are both practical and economical limits to this thickness. However, the life of chromium plated cylinders with heavy fuel and .8 mm (.0315 in.) of chromium

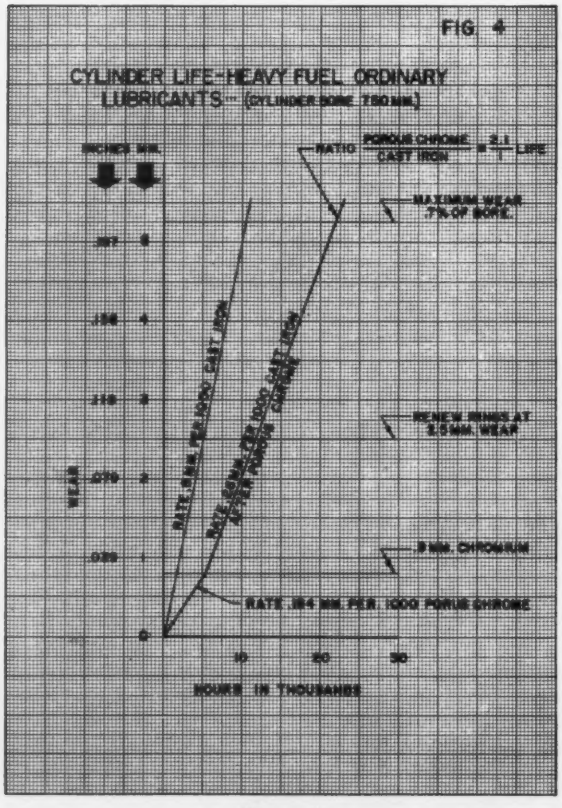


FIG. 4
CYLINDER LIFE-HEAVY FUEL ORDINARY LUBRICANTS - (CYLINDER BORE 750 MM.)
INCHES MM.
RATIO POROUS CHROME & CAST IRON
MAXIMUM WEAR .7% OF BORE
REWORK RINGS AT 2.5 MM. WEAR
8 MM. CHROMIUM
WEAR
HOURS IN THOUSANDS

Figure 4

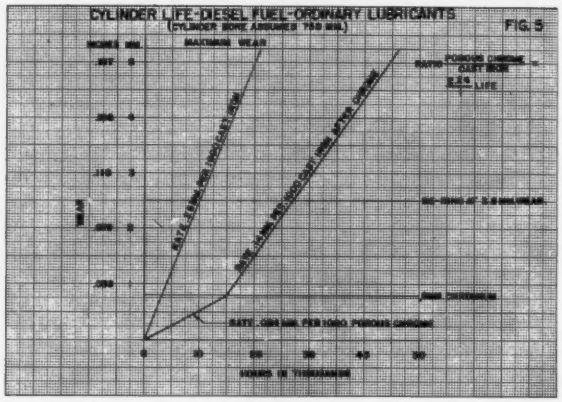


Figure 5

when worn to maximum limits (.7 per cent of the bore) has not shown a life of 4 to 1 because wear rates are, in most cases, accelerated when cast iron is reached. Figure 4 indicates that for maximum wear limit the life of the chromium cylinders is only about twice as great as cast iron and not 4 times greater as would be the case if all wear was on chromium. Figure 4 conditions are with heavy fuels and conventional lubricants. It is interesting to compare the two types of cylinders when operating on diesel fuels and conventional lubricants as shown in Figure 5. These data are also based upon ships listed in Tables 1 and 2 of reference (5). In this case, the wear rate on chromium only is 4½ times better than cast iron whereas it was 4 times better with heavy fuel, indicating some corrosive wear in the latter case. Particular interest is attached to the wear rates in cast iron after wearing through the chromium and the cast iron wear rates without the presence of any chromium in the cylinder. With diesel fuel the all cast iron cylinder wear rate is .25 mm (.0098 inch) per 1000 hours while the rate in cast iron after chromium is .14 mm (.0055 in.)* per 1000 hours or about ½ of normal.

WHY THIS IMPROVEMENT, WHICH HAS BEEN NOTED BY OTHERS?

Cylinder wear and ring wear are a continuing state of dimensional change wherein each is constantly endeavoring to accommodate, fit or match the other. A slow wear rate on one results in less wear for accommodation on the part of the other. By the same reasoning, slow wear on the remaining portion of chromium in the cylinder may account for a slower wear rate in the cast iron above it. When the chromium is just worn through at the top ring position the greater width of the ring is still riding on chromium. This situation continues successively with the lower rings as wear progresses into the cast iron and the chromium wearline creeps in the direction of bottom center. I have seen hun-

*See Appendix A for method of arriving at wear rate in cast iron after worn through chromium.

TABLE 2

Ship	Engine Type	Wear per 1000 hours Porus-Krome mm	Porus-Krome in.	Cast Iron mm	Cast Iron in.	Total Hours	Total Wear mm	Total Wear in.	Cylinder diameter mm	Cylinder diameter in.
3	2sda	.06	.0024			6,000	.4	.0157	620	24.409
12	2ssa	.03	.0012			24,000	.8	.0315	750	29.527
15	2sda	.05	.0020			14,000	.8	.0315	720	28.346
21	2sda	.07	.0028			10,000	.7	.0275	720	28.346
A8	2ssa	.10	.0039			2,500	.25	.0098	600	23.622
A12	2ssa	.02	.0008			24,000	.5	.0020	542	21.338
A1	2ssa	.05	.0020			20,000	1.0*	.0394	720	28.346
5	2ssa			.30	.0118	12,000	3.6	.1417	740	29.133
16	2sda			.30	.0118	10,000	3.0	.1181	720	28.346
22	2ssa			.32	.0126	10,000	3.25	.1279	750	29.527
A3	2ssa			.18	.0071	33,000	5.8	.2283	740	29.133
A12	2ssa			.13	.0051	24,000	3.1	.1220	542	21.338
A14	2sda			.3	.0120	15,000	4.6	.1811	600	23.622

* Just worn through .8 mm or .0315 in. chromium.

WEAR—HEAVY FUEL—STANDARD LUBRICANTS

Average wear rate: Porus-Krome .054 mm or .0021 in.
Cast Iron .25 mm or .098 in.

Wear ratio: Cast Iron : Porus-Krome = .25 mm : .054 mm = 4.6 : 1
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With heavy fuels the all cast iron cylinder wear rate is .63 mm (.0248 in.) per 1000 hours from Table 1. The literature mentions rates as low as .31 mm (.0122 in.) per 1000 hours[6] but considering many other sources an average wear rate of .5 mm (.020 in.) per 1000 hours seems representative and has been used in Figure 4. It is recognized that wear rates will even vary in different cylinders in the same engine but the approach here is to present a method and not a mathematical certainty.

Figure 6. 100 × Magnification.

On the basis of available data, wear curves have been plotted for marine diesel and heavy fuels with various lubricants. There will be disagreement with the various wear rates but an effort is made here to present a different approach to the problem of what conditions satisfy the best economy for the ship operator and the method is believed valid, regardless of the actual figures used. Ship owners and superintendent engineers may use the figures of their own experience but the general indications of the data will still apply.

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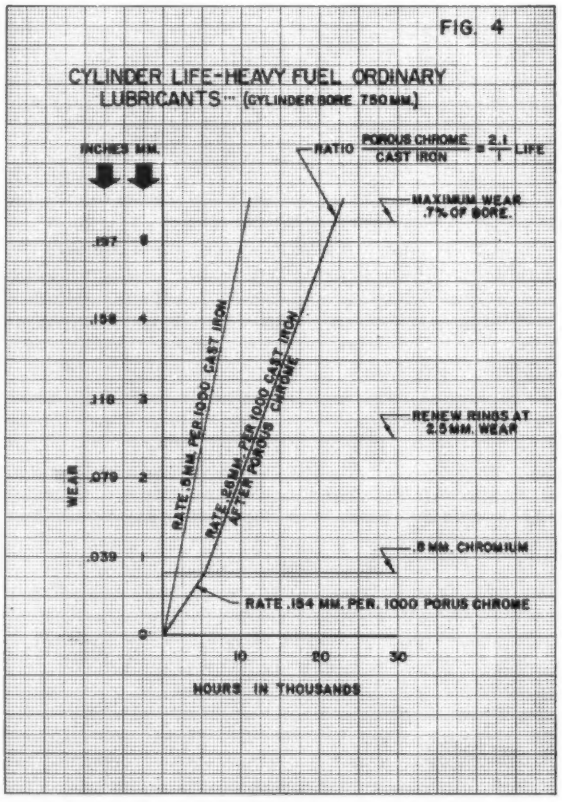

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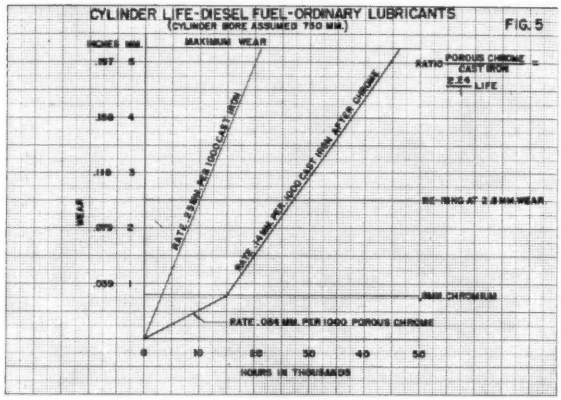

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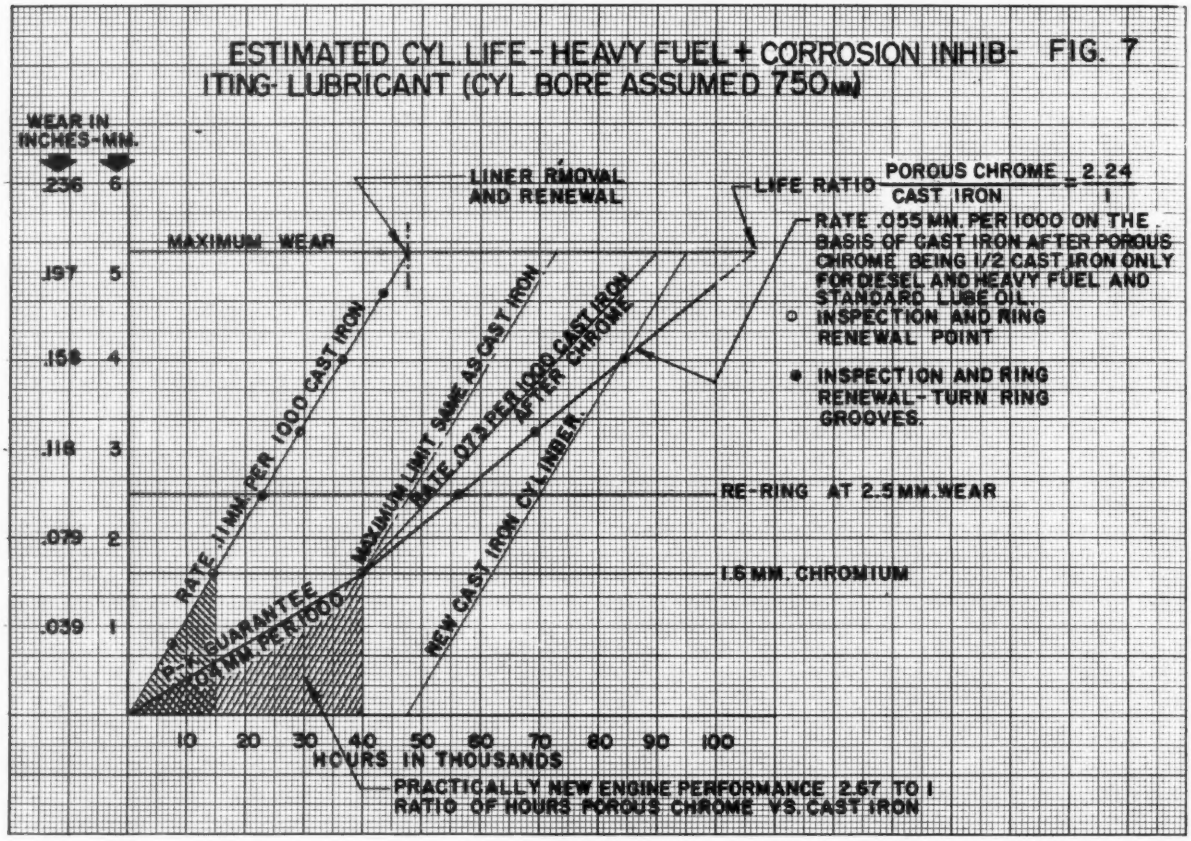

ESTIMATED CYL. LIFE - HEAVY FUEL + CORROSION INHIB- FIG. 7
ITING- LUBRICANT (CYL. BORE ASSUMED 750mm)
WEAR IN INCHES-MM.
.236 6
.197 5
.158 4
.118 3
.079 2
.039 1
LINER REMOVAL AND RENEWAL
MAXIMUM WEAR
LIFE RATIO POROUS CHROME / CAST IRON = 2.24 / 1
RATE .055 MM. PER 1000 ON THE BASIS OF CAST IRON AFTER POROUS CHROME BEING 1/2 CAST IRON ONLY FOR DIESEL AND HEAVY FUEL AND STANDARD LUBE OIL.
INSPECTION AND RING RENEWAL POINT
INSPECTION AND RING RENEWAL - TURN RING GROOVES.
RATE .11MM. PER 1000 CAST IRON
MAXIMUM LIMIT SAME AS CAST IRON
RATE .023 PER 1000 CAST IRON AFTER CHROME
RE-RING AT 2.5 MM. WEAR
1.6 MM. CHROMIUM
NEW CAST IRON CYLINDER
U.S. GUARANTEE .076 MM. PER 1000
10 20 30 40 50 60 70 80 90 100
HOURS IN THOUSANDS
PRACTICALLY NEW ENGINE PERFORMANCE 2.67 TO 1 RATIO OF HOURS POROUS CHROME VS. CAST IRON

Figure 7

ic optimum chromium thickness for any set of conditions, i.e., fuel used, lubricant and the established wear rates in the specific engine and service for chromium plated cylinders. Maintenance costs, inspection and ring replacements are influenced, not directly by the number of hours of operation but by the amount of wear which has taken place in that period, or putting it another way, by the cylinder wear rate. Some data from a large shipping company indicate that cylinder overhaul on a 740 mm (29.133 in.) engine is required at each .7-.8 mm (.0275-.0315 inch) wear and piston regrooving at around 2.5 mm (.098 inch) wear and this holds for both diesel and heavy fuels. On another type engine [7] serious ring breakage occurs at a total wear of 2½ to 3 mm (.098-.118 inch) when operating on heavy fuels. Both attrition wear and fatigue failure of rings rationally tie in with cylinder wall shape and the extent of cylinder taper which determines the stress range in which a piston ring operates. Further, a second set of rings should not be expected to last as long in the worn cylinder as did the original rings because the difference between the minimum and maximum cylinder diameters will increase with wear and it follows that the operating stress range in the ring increases. The fatigue limit for the larger stress range is lower than that for a small stress range,[8] therefore, the ring will fail earlier, i.e., at a lesser number of cycles. This raises the question of whether reduced maintenance should not call for reconsideration of the accepted wear limit of .7 per cent of the bore and lower this presently allowable limit. There are other factors, particularly fuel consumption, which point to economics of operation in lower allowable wear limits and savings in fuel cost which justify earlier removal of a cylinder from service.

Based upon cylinder overhaul required at every .8 mm (.0315 in.) wear and piston regrooving at 2.4 mm (.094 in.) wear we may make comparative costs of cast iron and chrome thicknesses of 1.1 mm (.043 in.), 1.6 mm (.062 in.) and 2.4 mm (.094 in.) at the wear rates shown on Figures 7 and 14. The following assumptions of costs will be made and are believed to be realistic although these will vary some with engines and shipyards. While costs are given in dollars and £ sterling,* they are not U. S. material and labor costs but European costs converted to dollars and £ sterling.

1. Cylinder overhaul and new piston rings..	$ 125.	44/12/10
2. Cylinder overhaul, new rings and turning ring grooves	175.	6/10/ 0

* Based on rate of exchange in 1954.

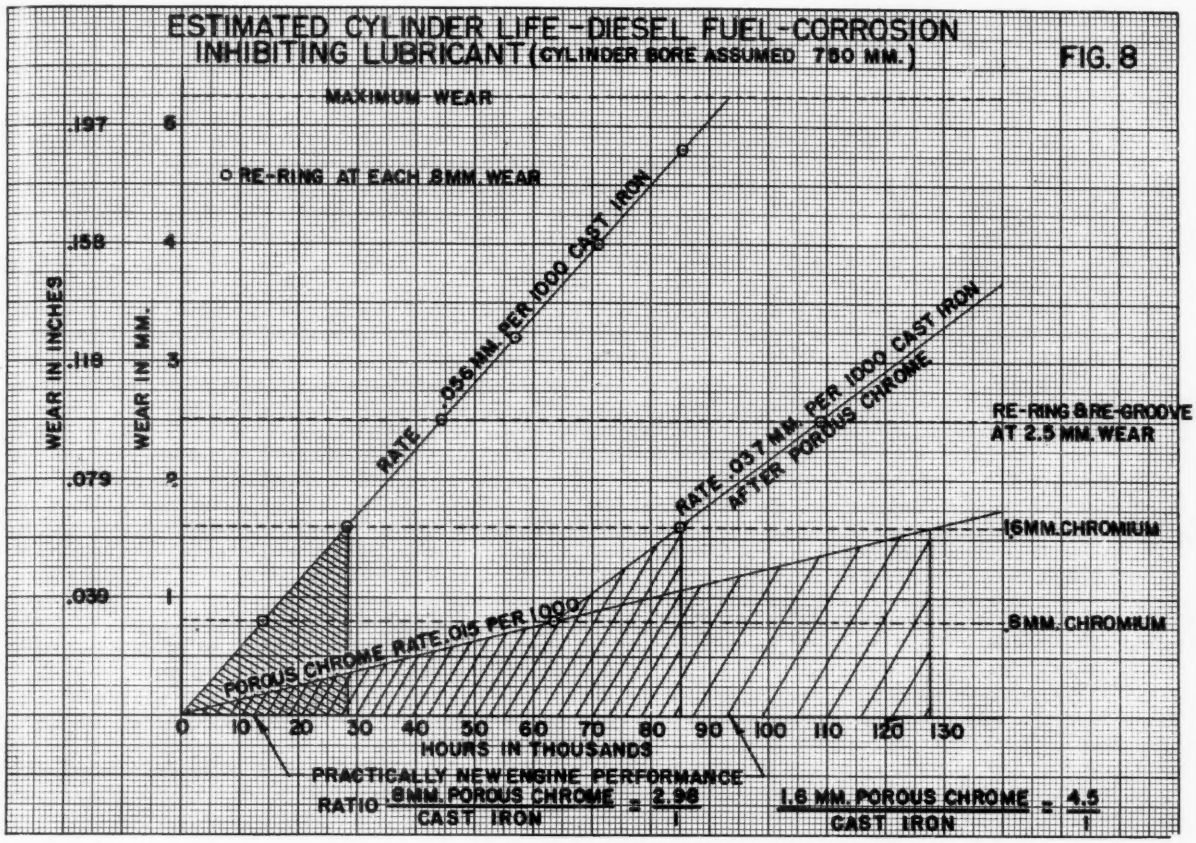

ESTIMATED CYLINDER LIFE - DIESEL FUEL - CORROSION INHIBITING LUBRICANT (CYLINDER BORE ASSUMED 750 MM.)
FIG. 8
MAXIMUM WEAR
RE-RING AT EACH 8MM. WEAR
RATE .056 MM. PER 1000 CAST IRON
RATE .037 MM. PER 1000 AFTER POROUS CHROME
RE-RING & RE-GROOVE AT 2.5 MM. WEAR
1.6MM. CHROMIUM
8MM. CHROMIUM
POROUS CHROME RATE .015 PER 1000
WEAR IN INCHES
WEAR IN MM.
.197
.158
.118
.079
.039
6
4
3
2
1
0 10 20 30 40 50 60 70 80 90 100 110 120 130
HOURS IN THOUSANDS
PRACTICALLY NEW ENGINE PERFORMANCE
RATIO 8MM. POROUS CHROME / CAST IRON = 2.96 / 1
1.6 MM. POROUS CHROME / CAST IRON = 4.5 / 1

Figure 8

3. Labor to remove and install a cylinder..	300.	107/ 2/10
4. Cost of a 740 mm (29.133 in.) cylinder, cast iron	1,640.	585/14/ 4
5. Cost of chrome plating 740 mm (29.133 in.) cylinder with 1.1 mm (.043 in.) chromium on the diameter	1,135.	405/ 7/ 0
6. Cost of 1.6 mm (.062 in.) chromium on the diameter	1,830.	653/11/ 5
7. Cost of chrome plating 740 mm (29.133 in.) cylinder with 2.4 mm (.094 in.) of chromium on the diameter	2,290.	817/17/ 2

The costs are calculated on the basis (Figures 7 and 14) of the life of the chrome plated cylinder being equal to that of two cast iron cylinders even though at this time there is still usable life remaining in the chrome plated cylinder.

For accumulated costs of two cast iron cylinders and maintenance up to 95,000 hours we have:

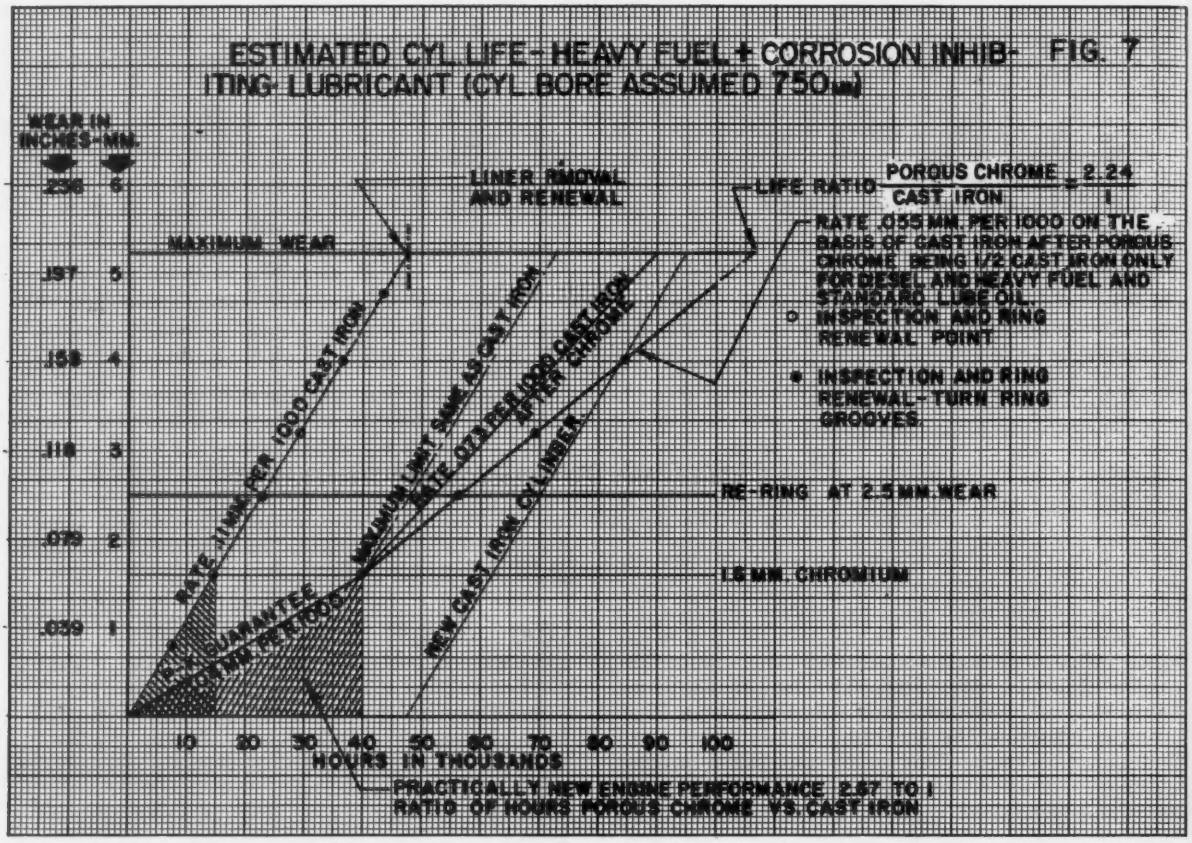
Cost of 2 cast iron cylinders at $1,640. (585/14/4)	$3,280.	1171/ 8/ 7
Cost of 8 inspections and ring renewals at $125. (44/12/10)	1,000.	357/ 2/10
Cost of 4 inspections, ring renewals and turning piston ring grooves	700.	250/ 0/ 0
Cost of removing and installing cylinders.	300.	107/ 2/10
Total accumulated cost	$5,280.	1885/14/ 3

For the accumulated cost of one chrome plated cylinder 1.1 mm (.043 in.) chromium and maintenance to 95,000 hours we have:

Cost of the cast iron cylinder	$1,640.	585/14/4
Cost of chrome plating 1.1 mm (.043 in.) on diameter	1,135.	405/7/1
Cost of 4 inspections and ring renewals...	500.	178/11/5
Cost of one inspection, ring renewal and turning grooves	175.	62/10/0
Total accumulated cost of 1 Porus-Krome cylinder 1.1 mm (.043 in.) on diameter	$3,450.	1232/2/10

For the accumulated costs of one chrome plated cylinder with 1.6 mm (.062 in.) of chromium on the diameter and maintenance to 95,000 hours we have:

Cost of the cast iron cylinder	$1,640.	585/14/4
Cost of chrome plating 1.6 mm (.062 in.) on diameter	1,830.	653/11/5
Cost of 4 inspections and ring renewals...	500.	178/11/5
Cost of one inspection, ring renewal and turning ring grooves	175.	62/10/0
Total accumulated costs of one Porus-Krome cylinder 1.6 mm (.062 in.) on diameter	$4,145.	1480/7/2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MAXIMUM WEAR
LIFE RATIO POROUS CHROME / CAST IRON = 2.24 / 1
RATE .055 MM PER 1000 ON THE BASIS OF CAST IRON AFTER POROUS CHROME BEING 1/2 CAST IRON ONLY FOR DIESEL AND HEAVY FUEL AND STANDARD LUBE OIL
INSPECTION AND RING RENEWAL POINT
INSPECTION AND RING RENEWAL - TURN RING GROOVES
RE-RING AT 2.5 MM. WEAR
1.6 MM. CHROMIUM
RATE .1 MM PER 1000 CAST IRON
NEW CAST IRON CYLINDER
10 20 30 40 50 60 70 80 90 100
HOURS IN THOUSANDS
PRACTICALLY NEW ENGINE PERFORMANCE 2.87 TO 1 RATIO OF HOURS POROUS CHROME VS. CAST IRON

Figure 7

ic optimum chromium thickness for any set of conditions, i.e., fuel used, lubricant and the established wear rates in the specific engine and service for chromium plated cylinders. Maintenance costs, inspection and ring replacements are influenced, not directly by the number of hours of operation but by the amount of wear which has taken place in that period, or putting it another way, by the cylinder wear rate. Some data from a large shipping company indicate that cylinder overhaul on a 740 mm (29.133 in.) engine is required at each .7-.8 mm (.0275-.0315 inch) wear and piston regrooving at around 2.5 mm (.098 inch) wear and this holds for both diesel and heavy fuels. On another type engine[7] serious ring breakage occurs at a total wear of 2½ to 3 mm (.098-.118 inch) when operating on heavy fuels. Both attrition wear and fatigue failure of rings rationally tie in with cylinder wall shape and the extent of cylinder taper which determines the stress range in which a piston ring operates. Further, a second set of rings should not be expected to last as long in the worn cylinder as did the original rings because the difference between the minimum and maximum cylinder diameters will increase with wear and it follows that the operating stress range in the ring increases. The fatigue limit for the larger stress range is lower than that for a small stress range,[8] therefore, the ring will fail earlier, i.e., at a lesser number of cycles. This raises the question of whether reduced maintenance should not call for reconsideration of the accepted wear limit of .7 per cent of the bore and lower this presently allowable limit. There are other factors, particularly fuel consumption, which point to economics of operation in lower allowable wear limits and savings in fuel cost which justify earlier removal of a cylinder from service.

Based upon cylinder overhaul required at every .8 mm (.0315 in.) wear and piston regrooving at 2.4 mm (.094 in.) wear we may make comparative costs of cast iron and chrome thicknesses of 1.1 mm (.043 in.), 1.6 mm (.062 in.) and 2.4 mm (.094 in.) at the wear rates shown on Figures 7 and 14. The following assumptions of costs will be made and are believed to be realistic although these will vary some with engines and shipyards. While costs are given in dollars and £ sterling,* they are not U. S. material and labor costs but European costs converted to dollars and £ sterling.

1. Cylinder overhaul and new piston rings..$ 125. 44/12/10
2. Cylinder overhaul, new rings and turning ring grooves 175. 6/10/ 0

* Based on rate of exchange in 1954.

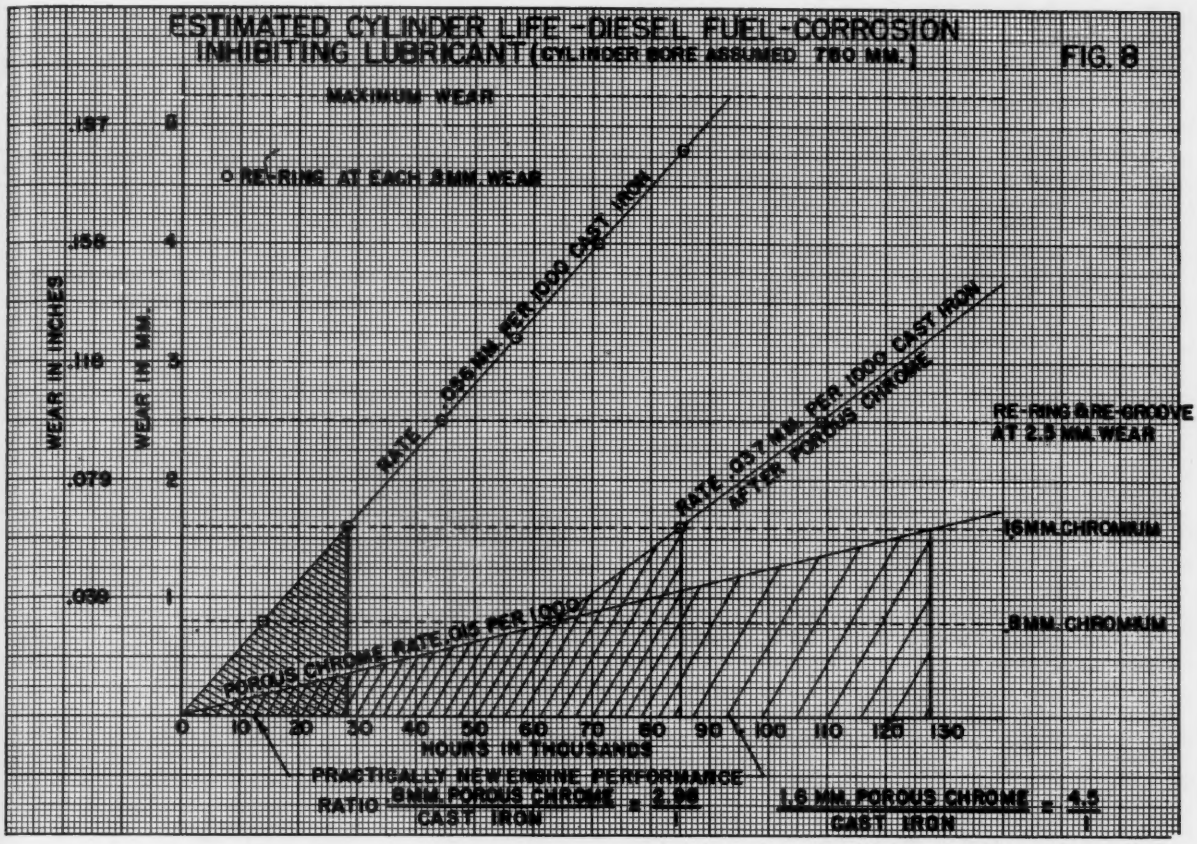
ESTIMATED CYLINDER LIFE - DIESEL FUEL - CORROSION INHIBITING LUBRICANT (CYLINDER BORE ASSUMED 760 MM.)
FIG. 8
MAXIMUM WEAR
RE-RING AT EACH 2 MM. WEAR
RATE .035 MM. PER 1000 CAST IRON
RATE .037 MM. PER 1000 CAST IRON AFTER POROUS CHROME
RE-RING & RE-GROOVE AT 2.5 MM. WEAR
1.5 MM. CHROMIUM
2 MM. CHROMIUM
POROUS CHROME RATE .015 PER 1000
WEAR IN INCHES
WEAR IN MM.
.197
.158
.118
.079
.039
5
4
3
2
1
0 10 20 30 40 50 60 70 80 90 100 110 120 130
HOURS IN THOUSANDS
PRACTICALLY NEW ENGINE PERFORMANCE
RATIO 2 MM. POROUS CHROME / CAST IRON = 2.28 / 1
1.6 MM. POROUS CHROME / CAST IRON = 4.5 / 1

Figure 8

3. Labor to remove and install a cylinder..	300.	107/ 2/10
4. Cost of a 740 mm (29.133 in.) cylinder, cast iron	1,640.	585/14/ 4
5. Cost of chrome plating 740 mm (29.133 in.) cylinder with 1.1 mm (.043 in.) chromium on the diameter	1,135.	405/ 7/ 0
6. Cost of 1.6 mm (.062 in.) chromium on the diameter	1,830.	653/11/ 5
7. Cost of chrome plating 740 mm (29.133 in.) cylinder with 2.4 mm (.094 in.) of chromium on the diameter	2,290.	817/17/ 2

The costs are calculated on the basis (Figures 7 and 14) of the life of the chrome plated cylinder being equal to that of two cast iron cylinders even though at this time there is still usable life remaining in the chrome plated cylinder.

For accumulated costs of two cast iron cylinders and maintenance up to 95,000 hours we have:

Cost of 2 cast iron cylinders at $1,640. (585/14/4)	$3,280.	1171/ 8/ 7
Cost of 8 inspections and ring renewals at $125. (44/12/10)	1,000.	357/ 2/10
Cost of 4 inspections, ring renewals and turning piston ring grooves	700.	250/ 0/ 0
Cost of removing and installing cylinders.	300.	107/ 2/10
Total accumulated cost	$5,280.	1885/14/ 3

For the accumulated cost of one chrome plated cylinder 1.1 mm (.043 in.) chromium and maintenance to 95,000 hours we have:

Cost of the cast iron cylinder	$1,640.	585/14/4
Cost of chrome plating 1.1 mm (.043 in.) on diameter	1,135.	405/7/1
Cost of 4 inspections and ring renewals...	500.	178/11/5
Cost of one inspection, ring renewal and turning grooves	175.	62/10/0
Total accumulated cost of 1 Porus-Krome cylinder 1.1 mm (.043 in.) on diameter	$3,450.	1232/2/10

For the accumulated costs of one chrome plated cylinder with 1.6 mm (.062 in.) of chromium on the diameter and maintenance to 95,000 hours we have:

Cost of the cast iron cylinder	$1,640.	585/14/4
Cost of chrome plating 1.6 mm (.062 in.) on diameter	1,830.	653/11/5
Cost of 4 inspections and ring renewals...	500.	178/11/5
Cost of one inspection, ring renewal and turning ring grooves	175.	62/10/0
Total accumulated costs of one Porus-Krome cylinder 1.6 mm (.062 in.) on diameter	$4,145.	1480/7/2

For the accumulated costs of one chrome plated cylinder with 2.4 mm (.094 in.) of chromium on the diameter and maintenance to 95,000 hours we have:

Cost of the cast iron cylinder	$1,640.	585/14/ 4
Cost of chrome plating 2.4 mm (.094 in.) on diameter	2,290.	817/17/ 2
Cost of 4 inspections and ring renewals	500.	178/11/ 5
Cost of 1 inspection, ring renewal and turning ring grooves	175.	62/10/ 0
Total accumulated cost of one Porus-Krome cylinder 2.4 mm (.094 in.) on diameter	$4,605.	1644/12/11

Summarizing, the indicated savings on initial costs plus maintenance over the period are:

For 1.1 mm (.043 in.) on dia.	$5280 minus $3450 = $1830 1855/14/3—1232/2/10=653/11/5
For 1.6 mm (.062 in.) on dia.	$5280 minus $4145 = $1135 1855/14/3—1480/7/2=405/7/1
For 2.4 mm (.094 in.) on dia.	$5280 minus $4605 = $ 675 1855/14/3—1644/12/11=241/1/5

or it may be said that plain iron cylinder operation cost respectively 53, 27 and 15 per cent more than a chrome plated cylinder.

It is notable that the 1.1 mm (.043 in.) chromium thickness represents savings in excess of the initial cost of a plain iron cylinder and maximum savings appear for this thin layer. This is due to the fact that maintenance costs are substantially the same for the several chromium thicknesses and the difference in savings lies in the initial cost of thicker chromium layers. However, initial and maintenance costs are only a part of the picture. They cannot be considered alone because the optimum economical chromium thickness is also influenced materially by fuel savings and other factors, all of which must be taken into account in combination.

The lower rate of wear associated with chromium involves a lesser amount of cylinder maintenance and conversely more certain engine performance. This is difficult to evaluate but certainly minimizes the possibility of ship immobilization. It may be averred that immobilization which can be an enormous cost figure cannot be charged against maintenance; nevertheless, extensive maintenance work may require more than normal port time. The best laid plans for such work can go awry and the more frequently that it has to be done, the greater the chances that it may be required at an inopportune time and result in temporary unscheduled ship immobilization.

FUEL COSTS VS EXTENT OF WEAR

The slow rate of wear of Porus-Krome cylinders has the favorable advantage of keeping the engine operating a longer time in the near new condition as concerns cylinder dimensions and thus maintaining overall engine performance. Cylinder wear is reflected in:

Loss of compression pressure
Increase in specific fuel consumption
Increase in lube oil consumption

and these factors are not linearly proportional to wear, but increase at a faster rate as wear progresses. The trend of these factors is strikingly shown in Report Project Number 1-237 of the Armour Research Foundation, Illinois Institute of Technology. While these tests were run on a 4-cycle engine at 1000 rpm, the general trend of these curves should hold for large two-cycle slow speed engines. To make the data more easily interpreted, it has been replotted against percentage of cylinder bore as shown on Figures 9 and 10.

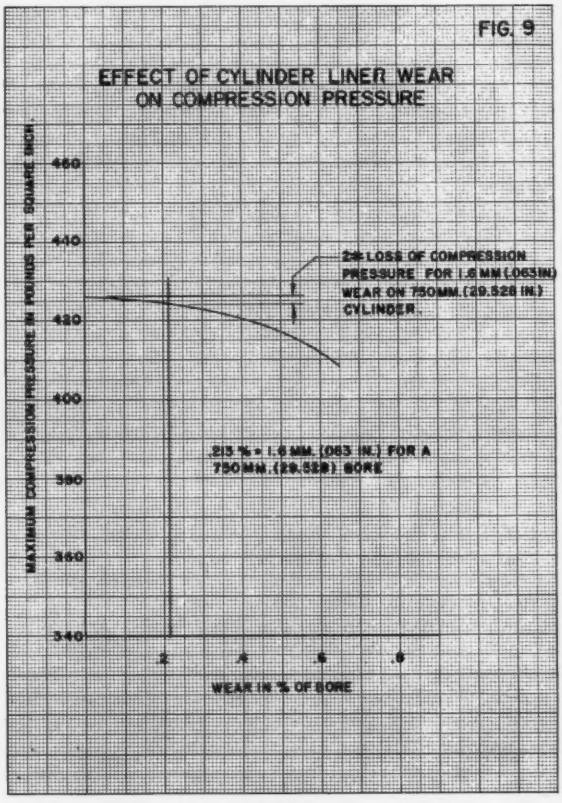
FIG. 9
EFFECT OF CYLINDER LINER WEAR
ON COMPRESSION PRESSURE
MAXIMUM COMPRESSION PRESSURE IN POUNDS PER SQUARE INCH
460
440
420
400
380
360
340
2 4 6 8
WEAR IN % OF BORE
2# LOSS OF COMPRESSION PRESSURE FOR 1.6 MM (.063 IN.) WEAR ON 750 MM (29.528 IN.) CYLINDER.
.213 % = 1.6 MM (.063 IN.) FOR A 750 MM (29.528) BORE

Figure 9

Figure 9 shows rapidly increasing loss of compression with wear but at 1.6 mm or .062 in. (.213 per cent of a 750 mm or 29.527 in. bore) chrome thickness, the loss in compression is only 2 lbs., which is practically new engine performance. Reference to curve No. 7 indicates this nearly new engine performance can be expected to persist for 40,000 hours with the chrome plated cylinder as against 15,000 hours for a cast iron cylinder. On the other hand, at a wear of .7 per cent (5.25 mm. or .020 in. for a 750 mm or 29.527 in. bore) the loss in compression was 20 lbs., at which point experience

TABLE 3

1	2	3		4		5		6	7		8	9	
Ship	Engine Type	Wear per 1000 hours— Combined Porus Krome and Cast Iron				Total Wear Porus Krome + Cast Iron		Total Hours	Total Wear in Column 5 minus .8 mm (.0315 in.) of chromium = wear into Cast Iron		Total Hours in Column 6 minus 14,800 hours for diesel = hours running on cast iron OR Total Hours in Column 6 minus 5,200 hours for heavy fuel = hours running on cast iron*	Wear Rate in Cast Iron after worn through chromium = Column 7 / Column 8	
		Diesel Fuel		Heavy Fuel								mm / 1000 hrs.	in. / 1000 hrs.
		mm	in.	mm	in.	mm	in.		mm	in.		mm	in.
6	2ssa	.08	.0031			1.5	.059	18,000	.7	.028	3,200	.218	.0085
11	2sda	.12	.0047			2.4	.094	19,000	1.6	.062	4,200	.38	.0150
A1	2ssa	.05	.0020			1.0	.039	20,000	.2	.007	5,200	.038	.0015
A2	2ssa	.11	.0043			2.45	.096	27,000	1.65	.064	12,200	.135	.0053
A5	2ssa	.05	.0020			1.1	.043	22,000	.3	.012	7,200	.042	.0016
A13	2sda	.04	.0016			1.4	.055	36,000	.6	.024	21,200	.028	.0011
A14	2sda	.09	.0035			2.3	.090	25,400	1.5	.059	10,600	.14	.0055
AVERAGE		.077	.0030					27,060			9,100	.14	.0055
1	2ssa			.09	.0035	1.5	.059	17,000	.7	.028	11,800	.059	.0023
4	2sda			.23	.0091	2.5	.098	12,000	1.7	.066	6,800	.25	.0098
7	2ssa			.13	.0051	1.3	.051	10,000	.5	.020	4,800	.104	.0040
13	2ssa			.16	.0063	2.5	.098	15,000	1.7	.066	10,800	.157	.0061
15	2sda			.32	.0126	3.2	.125	10,000	2.4	.094	4,800	.50	.0197
18	2sda			.35	.0138	3.5	.137	10,000	2.7	.010	4,800	.56	.0220
A1	2ssa			.13	.0051	1.2	.047	9,000	.4	.016	3,800	.105	.0041
A6	2ssa			.22	.0087	2.3	.090	10,500	1.5	.059	5,300	.284	.0111
A10	2ssa			.22	.0087	2.0	.078	9,000	1.2	.047	3,800	.316	.0124
AVERAGE				.21	.0083			11,400			6,300	.26	.0102

TABLE OF SHIPS WORN THROUGH CHROMIUM INTO CAST IRON—AVERAGE WEAR ON COMBINATION

*Derived as follows:

$$\text{For diesel fuel} — \frac{\text{.8mm (.0315 in.) Porus Krome}}{\text{Average of .054 mm (.0212 in.) per 1000 hrs. from Table 2}} = \text{14,800 hours life of Porus Krome on diesel fuel}$$

$$\text{For heavy fuel} — \frac{\text{.8mm (.0315 in.) Porus Krome}}{\text{Average of .054 mm (.0212 in.) per 1000 hrs. from Table 1}} = \text{5,200 hours life of Porus Krome on heavy fuel}$$

indicates excessive blowby, loss of power and misfiring. The fast increasing rate of pressure loss beyond this point is experimental confirmation of what has been learned in practice, that wear beyond a certain point makes engine operation painful and uneconomical, if not impossible.

The effect of lowered compression pressure on thermal efficiency is shown on Curve No. 10. Considering .7 per cent of bore as top wear limit, it can be seen that an original engine $1.00 (7s 1d) worth of horsepower is now costing $1.085 (7s 9d) and this does not include the laundry bill from dirty exhaust.

At 1.6 mm (.062 in.) of chrome wear the indicated increase in cost after 40,000 hours is only $.015 (1d).

The tremendous effect of wear on lubricating oil

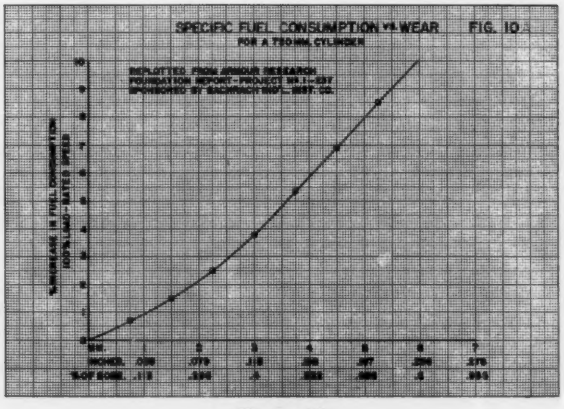

Figure 10

For the accumulated costs of one chrome plated cylinder with 2.4 mm (.094 in.) of chromium on the diameter and maintenance to 95,000 hours we have:

Cost of the cast iron cylinder	$1,640.	585/14/4
Cost of chrome plating 2.4 mm (.094 in.) on diameter	2,290.	817/17/2
Cost of 4 inspections and ring renewals	500.	178/11/5
Cost of 1 inspection, ring renewal and turning ring grooves	175.	62/10/0
Total accumulated cost of one Porus-Krome cylinder 2.4 mm (.094 in.) on diameter	$1,605.	1644/12/11

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For 1.1 mm (.043 in.) on dia.	$5280 minus $3450 = $1830 1855/14/3—1232/2/10=653/11/5
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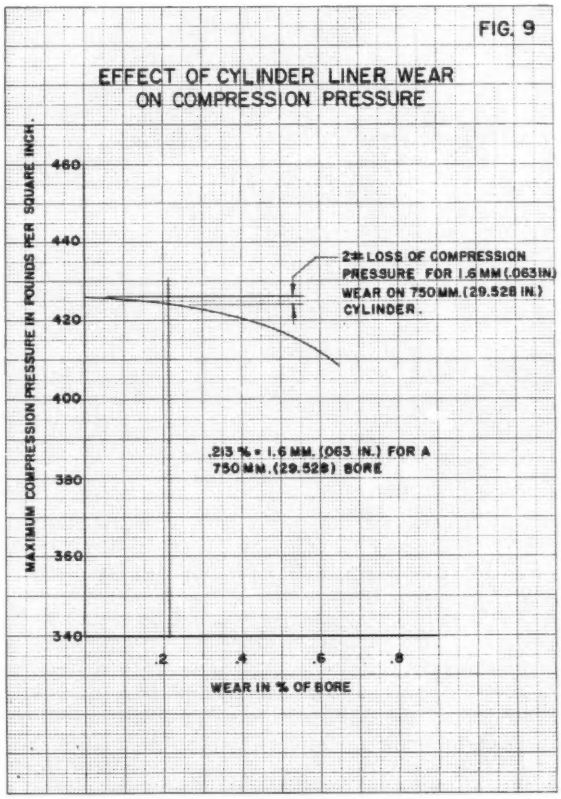

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		Diesel Fuel		Heavy Fuel									
		mm	in.	mm	in.	mm	in.		mm	in.		mm	in.
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A2	2ssa	.11	.0043			2.45	.096	27,000	1.65	.064	12,200	.135	.0053
A5	2ssa	.05	.0020			1.1	.043	22,000	.3	.012	7,200	.042	.0016
A13	2sda	.04	.0016			1.4	.055	36,000	.6	.024	21,200	.028	.0011
A14	2sda	.09	.0035			2.3	.090	25,400	1.5	.059	10,600	.14	.0055
AVERAGE		.077	.0030					27,060			9,100	.14	.0055
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4	2sda			.23	.0091	2.5	.098	12,000	1.7	.066	6,800	.25	.0098
7	2ssa			.13	.0051	1.3	.051	10,000	.5	.020	4,800	.104	.0040
13	2ssa			.16	.0063	2.5	.098	15,000	1.7	.068	10,800	.157	.0061
15	2sda			.32	.0126	3.2	.125	10,000	2.4	.094	4,800	.50	.0197
18	2sda			.35	.0138	3.5	.137	10,000	2.7	.010	4,800	.56	.0220
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indicates excessive blowby, loss of power and misfiring. The fast increasing rate of pressure loss beyond this point is experimental confirmation of what has been learned in practice, that wear beyond a certain point makes engine operation painful and uneconomical, if not impossible.

The effect of lowered compression pressure on thermal efficiency is shown on Curve No. 10. Considering .7 per cent of bore as top wear limit, it can be seen that an original engine $1.00 (7s 1d) worth of horsepower is now costing $1.085 (7s 9d) and this does not include the laundry bill from dirty exhaust.

At 1.6 mm (.062 in.) of chrome wear the indicated increase in cost after 40,000 hours is only $.015 (1d).

The tremendous effect of wear on lubricating oil

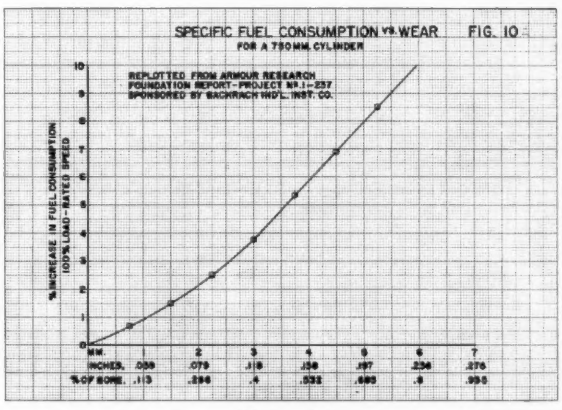
SPECIFIC FUEL CONSUMPTION VS WEAR
FOR A 750 MM. CYLINDER
FIG. 10
REPLOTTED FROM ARMOUR RESEARCH FOUNDATION REPORT—PROJECT NR. 1-237 SPONSORED BY BACKRACH IND'L. INST. CO.
% INCREASE IN FUEL CONSUMPTION 100% LOAD - RATED SPEED
MM.
INCHES
% OF BORE

Figure 10

consumption is shown on Curve No. 11. While it is true this particular engine was splash lubricated, the result is due to inability of rings to control oil scraping on taper worn cylinder walls and burning of the oil film by hot gas blowby. The situation will not be nearly as acute on a large ship engine with individual lubricators; nevertheless, some of the effects will carry over and excessive wear will occur with blowby.

A major cost of cylinder wear is reflected in the resulting increase in specific fuel consumption (Figure 10). Using Figure 10 and wear curves such as Figures 7 and 14, the increasing cost of fuel vs wear or hours may be computed. This has been done in Tables 4, 5 and 6 for the conditions of cast iron and various thicknesses of chromium operating on heavy fuel with a corrosion inhibiting lubricant. Similar tables could be prepared for any set of conditions such as shown on Figure 5 for operation on diesel fuels and ordinary lubricants. The tables are computed on the basis of:

1000 brake horsepower per cylinder
specific normal fuel consumption of .36 lbs. per BHP hr.
fuel cost of 023.50 (8/7/10) per long ton.

The data from the tables are plotted on Figure 13 and strikingly show that the chromium plated cylinders are saving fuel dollars/pounds—until the partly worn chromium cylinder has to compete with a new cast iron cylinder in which there is no wear. The accumulated savings on the chromium cylinder now decrease until the total wear on the

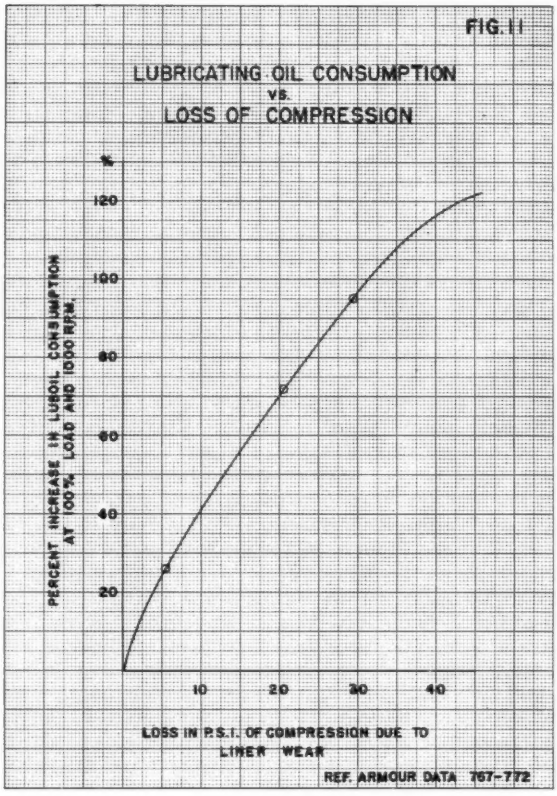

Figure 11

TABLE 4

10,000 hours period	Wear in Cast Iron	Percent increase in fuel cons.			Wear in Porus-Krome	Percent increase in fuel cons.			Percent diff. in average Cast Iron	$ saving for period	accumulated saving
	mm	begin	end	average	mm	begin	end	average	Porus-Krome		
1	1.1	0	1.1	.55	.4	0	.4	.2	.35	132	132
2	2.2	1.1	2.25	1.67	.8	.4	.8	.6	1.07	404	536
3	3.3	2.25	4.4	3.32	1.2	.8	1.2	1.0	2.32	877	1413
4	4.4	4.4	6.7	5.55	1.6	1.2	1.6	1.4	4.15	1570	2983
4-4.75	5.25	6.7	8.6	7.65	2.0	1.6	2.2	1.9	5.75	2175	5158
4.75-5.75	1.1	0	1.1	0.55	2.55	2.2	3.	2.6	—2.05	—775	4383
5.75-6.75	2.2	1.1	2.25	1.67	3.1	3.	4.	3.5	—1.83	—690	3693
6.75-7.75	3.3	2.25	4.4	3.32	3.65	4.	5.1	4.5	—1.18	—445	3248
7.75-8.75	4.4	4.4	6.7	5.05	4.2	5.1	6.2	5.6	— .55	—208	3040
8.75-9.5	5.25	6.7	8.6	7.65	4.6	6.2	7.1	6.6	+1.05	+397	3437

Saving for period = Average percent difference × standard fuel cost

Heavy Fuels — Corrosion Inhibiting Lubricants — 1.6 mm chromium

TABLE 5

10,000 hours period	Wear in Cast Iron	Percent increase in fuel cons.			Wear in Porus-Krome	Percent increase in fuel cons.			Percent diff. in average Cast Iron : Porus-Krome	$ saving for period	accumulated saving
	mm	begin	end	average	mm	begin	end	average			
1	1.1	0	1.1	.55	.4	0	.4	.2	.35	133	132
2	2.2	1.1	2.25	1.67	.8	.4	.8	.6	1.07	404	536
2-2.75	3.05	2.25	3.4	2.28	1.1	.8	1.1	.95	1.87	707	1243
2.75-3.75	4.15	3.4	5.3	4.35	1.67	1.1	1.7	1.4	2.95	1120	2363
3.75-4.75	5.25	5.3	7.5	6.4	2.25	1.7	2.5	2.1	4.3	1630	3993
4.75-5.75	1.1	0	1.1	.55	2.8	2.5	3.5	3.0	—2.45	—927	3066
5.75-6.75	2.2	1.1	2.25	1.67	3.35	3.5	4.5	4.0	—2.33	—880	2186
6.75-7.75	3.3	2.25	4.4	3.32	3.9	4.5	5.6	5.05	11.73	—654	1532
7.75-8.75	4.4	4.4	6.7	5.55	4.45	5.6	6.8	6.2	— .65	—245	1287
8.75-9.5	5.25	6.7	8.6	7.65	4.85	6.8	7.7	7.2	+ .45	+170	1457

Heavy Fuels — Corrosion Inhibiting Lubricants — 1.1 mm chromium

Saving for period = Average percent difference × standard fuel cost

TABLE 6

10,000 hours period	Wear in Cast Iron	Percent increase in fuel cons.			Wear in Porus-Krome	Percent increase in fuel cons.			Percent diff. in average Cast Iron : Porus-Krome	$ saving for period	accumulated saving
	mm	begin	end	average	mm	begin	end	average			
1	1.1	0	1.1	.55	.4	0	.4	.2	.35	132	132
2	2.2	1.1	2.25	1.67	.8	.4	.8	.6	1.07	404	536
3	3.3	2.25	4.4	3.32	1.2	.8	1.2	1.0	2.32	877	1413
4	4.4	4.4	6.7	5.55	1.6	1.2	1.6	1.4	4.15	1570	2983
4-4.75	5.25	6.7	8.6	7.65	1.9	1.6	2.1	1.85	5.80	2190	5173
4.75-5.75	1.1	0	1.1	.55	2.3	2.1	2.6	2.35	—1.80	—680	4493
5.75-6.75	2.2	1.1	2.25	1.67	2.85	2.6	3.55	3.07	—1.40	—528	3965
6.75-7.75	3.3	2.25	4.4	3.32	3.4	3.55	4.7	4.12	— .80	—320	3645
7.75-8.75	4.4	4.4	6.7	5.55	3.9	4.7	5.6	5.15	+ .40	+151	3796
8.75-9.5	5.25	6.7	8.6	7.65	4.35	5.6	6.6	6.1	+1.55	+585	4381

Heavy Fuels — Corrosion Inhibiting Lubricants — 2.4 mm chromium

Saving for period = Average percent difference × standard fuel cost

consumption is shown on Curve No. 11. While it is true this particular engine was splash lubricated, the result is due to inability of rings to control oil scraping on taper worn cylinder walls and burning of the oil film by hot gas blowby. The situation will not be nearly as acute on a large ship engine with individual lubricators; nevertheless, some of the effects will carry over and excessive wear will occur with blowby.

A major cost of cylinder wear is reflected in the resulting increase in specific fuel consumption (Figure 10). Using Figure 10 and wear curves such as Figures 7 and 14, the increasing cost of fuel vs wear or hours may be computed. This has been done in Tables 4, 5 and 6 for the conditions of cast iron and various thicknesses of chromium operating on heavy fuel with a corrosion inhibiting lubricant. Similar tables could be prepared for any set of conditions such as shown on Figure 5 for operation on diesel fuels and ordinary lubricants. The tables are computed on the basis of:

1000 brake horsepower per cylinder
specific normal fuel consumption of .36 lbs. per BHP hr.
fuel cost of 023.50 (8/7/10) per long ton.

The data from the tables are plotted on Figure 13 and strikingly show that the chromium plated cylinders are saving fuel dollars/pounds—until the partly worn chromium cylinder has to compete with a new cast iron cylinder in which there is no wear. The accumulated savings on the chromium cylinder now decrease until the total wear on the

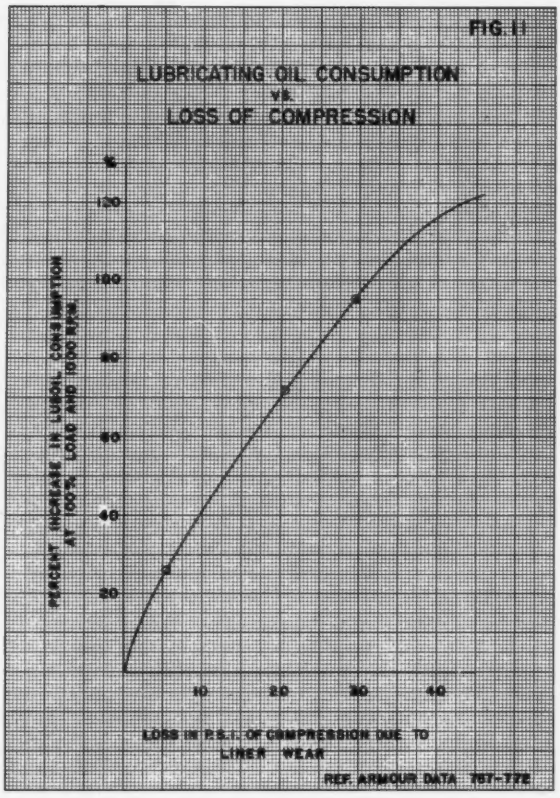

Figure 11

TABLE 4

10,000 hours period	Wear in Cast Iron	Percent increase in fuel cons.			Wear in Porus-Krome	Percent increase in fuel cons.			Percent diff. in average Cast Iron : Porus-Krome	$ saving for period	accumulated saving
	mm	begin	end	average	mm	begin	end	average			
1	1.1	0	1.1	.55	.4	0	.4	.2	.35	132	132
2	2.2	1.1	2.25	1.67	.8	.4	.8	.6	1.07	404	536
3	3.3	2.25	4.4	3.32	1.2	.8	1.2	1.0	2.32	877	1413
4	4.4	4.4	6.7	5.55	1.6	1.2	1.6	1.4	4.15	1570	2983
4-4.75	5.25	6.7	8.6	7.65	2.0	1.6	2.2	1.9	5.75	2175	5158
4.75-5.75	1.1	0	1.1	0.55	2.55	2.2	3.	2.6	—2.05	—775	4383
5.75-6.75	2.2	1.1	2.25	1.67	3.1	3.	4.	3.5	—1.83	—690	3693
6.75-7.75	3.3	2.25	4.4	3.32	3.65	4.	5.1	4.5	—1.18	—445	3248
7.75-8.75	4.4	4.4	6.7	5.05	4.2	5.1	6.2	5.6	— .55	—208	3040
8.75-9.5	5.25	6.7	8.6	7.65	4.6	6.2	7.1	6.6	+1.05	+397	3437

Saving for period = Average percent difference × standard fuel cost

Heavy Fuels — Corrosion Inhibiting Lubricants — 1.6 mm chromium

TABLE 5

10,000 hours period	Wear in Cast Iron	Percent increase in fuel cons.			Wear in Porus-Krome	Percent increase in fuel cons.			Percent diff. in average Cast Iron : Porus-Krome	$ saving for period	accumulated saving
	mm	begin	end	average	mm	begin	end	average			
1	1.1	0	1.1	.55	.4	0	.4	.2	.35	132	132
2	2.2	1.1	2.25	1.67	.8	.4	.8	.6	1.07	404	536
2-2.75	3.05	2.25	3.4	2.28	1.1	.8	1.1	.95	1.87	707	1243
2.75-3.75	4.15	3.4	5.3	4.35	1.67	1.1	1.7	1.4	2.95	1120	2363
3.75-4.75	5.25	5.3	7.5	6.4	2.25	1.7	2.5	2.1	4.3	1630	3993
4.75-5.75	1.1	0	1.1	.55	2.8	2.5	3.5	3.0	—2.45	—927	3066
5.75-6.75	2.2	1.1	2.25	1.67	3.35	3.5	4.5	4.0	—2.33	—880	2186
6.75-7.75	3.3	2.25	4.4	3.32	3.9	4.5	5.6	5.05	11.73	—654	1532
7.75-8.75	4.4	4.4	6.7	5.55	4.45	5.6	6.8	6.2	— .65	—245	1287
8.75-9.5	5.25	6.7	8.6	7.65	4.85	6.8	7.7	7.2	+ .45	+170	1457

Heavy Fuels — Corrosion Inhibiting Lubricants — 1.1 mm chromium

Saving for period = Average percent difference × standard fuel cost

TABLE 6

10,000 hours period	Wear in Cast Iron	Percent increase in fuel cons.			Wear in Porus-Krome	Percent increase in fuel cons.			Percent diff. in average Cast Iron : Porus-Krome	$ saving for period	accumulated saving
	mm	begin	end	average	mm	begin	end	average			
1	1.1	0	1.1	.55	.4	0	.4	.2	.35	132	132
2	2.2	1.1	2.25	1.67	.8	.4	.8	.6	1.07	404	536
3	3.3	2.25	4.4	3.32	1.2	.8	1.2	1.0	2.32	877	1413
4	4.4	4.4	6.7	5.55	1.6	1.2	1.6	1.4	4.15	1570	2983
4-4.75	5.25	6.7	8.6	7.65	1.9	1.6	2.1	1.85	5.80	2190	5173
4.75-5.75	1.1	0	1.1	.55	2.3	2.1	2.6	2.35	—1.80	—680	4493
5.75-6.75	2.2	1.1	2.25	1.67	2.85	2.6	3.55	3.07	—1.40	—528	3965
6.75-7.75	3.3	2.25	4.4	3.32	3.4	3.55	4.7	4.12	— .80	—320	3645
7.75-8.75	4.4	4.4	6.7	5.55	3.9	4.7	5.6	5.15	+ .40	+151	3796
8.75-9.5	5.25	6.7	8.6	7.65	4.35	5.6	6.6	6.1	+1.55	+585	4331

Heavy Fuels — Corrosion Inhibiting Lubricants — 2.4 mm chromium

Saving for period = Average percent difference × standard fuel cost

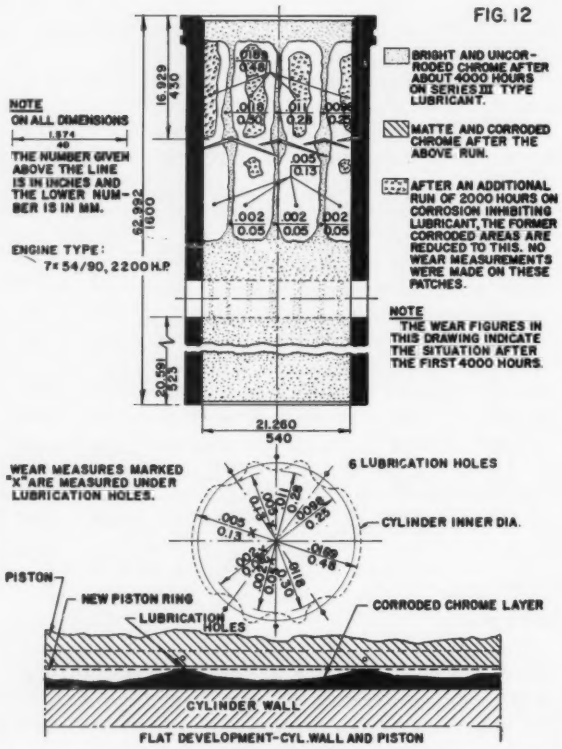
FIG. 12
BRIGHT AND UNCORRODED CHROME AFTER ABOUT 4000 HOURS ON SERIES III TYPE LUBRICANT.
MATTE AND CORRODED CHROME AFTER THE ABOVE RUN.
AFTER AN ADDITIONAL RUN OF 2000 HOURS ON CORROSION INHIBITING LUBRICANT, THE FORMER CORRODED AREAS ARE REDUCED TO THIS. NO WEAR MEASUREMENTS WERE MADE ON THESE PATCHES.
NOTE
THE WEAR FIGURES IN THIS DRAWING INDICATE THE SITUATION AFTER THE FIRST 4000 HOURS.
NOTE
ON ALL DIMENSIONS
THE NUMBER GIVEN ABOVE THE LINE IS IN INCHES AND THE LOWER NUMBER IS IN MM.
ENGINE TYPE:
7-54/90, 2200 H.P.
16.929
430
62.992
1600
20.591
523
21.260
540
WEAR MEASURES MARKED "X" ARE MEASURED UNDER LUBRICATION HOLES.
6 LUBRICATION HOLES
CYLINDER INNER DIA.
PISTON
NEW PISTON RING
LUBRICATION HOLES
CORRODED CHROME LAYER
CYLINDER WALL
FLAT DEVELOPMENT-CYL. WALL AND PISTON

Figure 12

second cast iron cylinder reaches that on the original chrominum cylinder which occurs around period 8.25-8.75.

Here the chromium plated cylinder begins to make savings again and get back some of its lost profit. In spite of the losing period the plated cylinders all end up with substantial savings. BUT—would a smart accountant or the manager of an investment fund be satisfied with a $1500 (535/14/3) profit (1.1 mm or .043 in. chromium) in 9½ periods when he could have had a profit of $4000 (1428/11/5) in one-half the time? He would not be happy and neither should a ship operator. The right move is simple. The chromium plated cylinder should be removed from service at the point where a cast iron cylinder would have reached its maximum wear limit, i.e., at period 4.75. At this point the plating cost has been earned and the maximum savings have accrued. It should be replaced with another chromium plated cylinder to start earning savings all over again from 0 to 4.75 period on Figure 13. Reference to Figure 7 and 14 will show that the worn chromium plated cylinder has monetary salvage value which may be nearly equal to the price of a new cast iron cylinder. The wear into the cast iron through the chromium is 1.1 mm (.043 in.) for the 1.1 mm (.043 in.) thick chromium; .4 mm (.0157 in.) for the 1.6 mm (.062 in.) thick chromium and 0 for the 2.4 mm (.094 in.) thick chromium.

For a cost slightly in excess of the original plating costs, the cylinder can be stripped of its remaining chromium, honed to a straight round bore and replated back to the original standard bore dimension with chromium or electrolytic iron and chromium.

The question is not yet answered as to what is the optimum economic chromium thickness for this set of conditions. To arrive at this we must consider together fuel savings, original costs and maintenance savings, all at situations corresponding to the life of one cast iron cylinder and also two cast iron cylinders. The capital and maintenance costs have been set down for the two cast iron cylinder life and from these costs they can be computed for the life of one cast iron cylinder. Similarly, fuel savings can be taken from Tables 4, 5 and 6.

The combined savings for both 1 and 2 cylinder life are plotted on Figure 15. In no case is the combined savings of the 2 cylinder life curve twice as great as the one cylinder life savings, yet it has taken twice as long. It follows that the maximum rate of earning lies in the one cylinder life curve and the optimum economic chromium thickness is

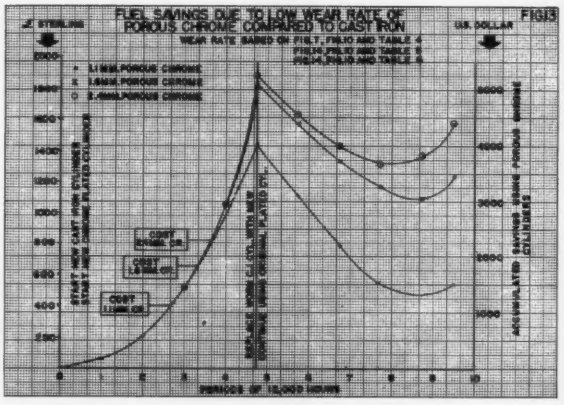
FUEL SAVINGS DUE TO LOW WEAR RATE OF POROUS CHROME COMPARED TO CAST IRON
FIG. 13
£ STERLING
U.S. DOLLAR
WEAR RATE BASED ON FIG. 7, FIG. 10 AND TABLE 4
FIG. 14, FIG. 10 AND TABLE 5
FIG. 14, FIG. 10 AND TABLE 6
1.1 MM. POROUS CHROME
1.6 MM. POROUS CHROME
2.4 MM. POROUS CHROME
START NEW CAST IRON CYLINDER
START NEW CHROME PLATED CYLINDER
COST ENGINE OR.
COST 1.1 MM. CR.
COST 1.6 MM. CR.
REPLACE WORN CAST IRON CYLINDER
REPLACE WORN CHROME PLATED CYLINDER
ACCUMULATED SAVINGS FROM POROUS CHROME CYLINDERS
PERIODS OF 1,000 HOURS

Figure 13

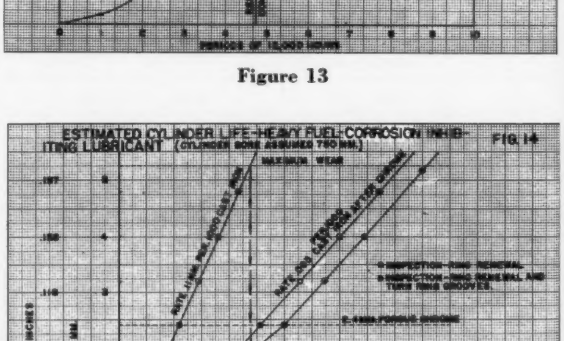
ESTIMATED CYLINDER LIFE-HEAVY FUEL-CORROSION INHIBITING LUBRICANT (CYLINDER BORE ASSUMED 760 MM.)
FIG. 14
MAXIMUM WEAR
1.1 MM. POROUS CHROME
WEAR IN INCHES
WEAR IN MM.
THOUSANDS OF HOURS

Figure 14

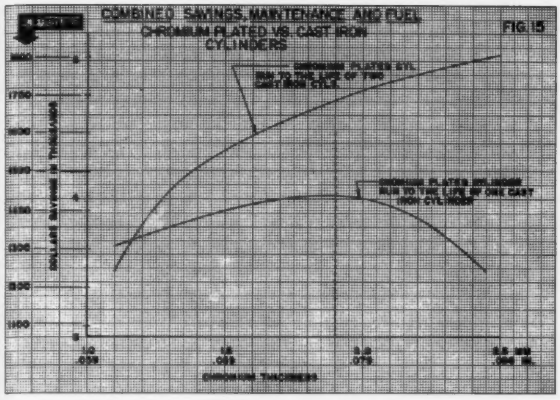

Figure 15

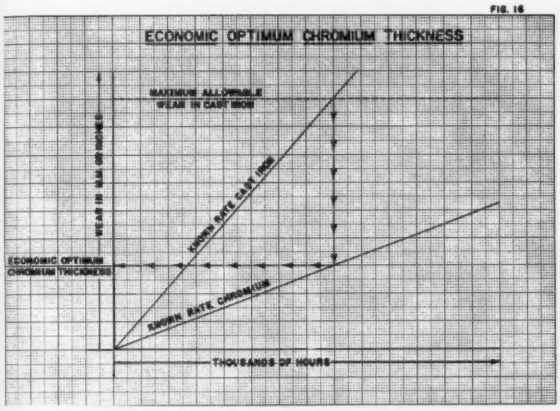

Figure 16

at the peak of the curve for this particular case, at a chromium thickness of 1.90 mm (.074 in.).

As a corollary to this detailed procedure there is a simpler way of establishing the economic chromium thickness for any set of conditions, i.e., any fuel, any type of service or any lubricant, providing the wear rates of cast iron and chromium only are known for that set of conditions. The method follows (see Figure 16):

(1) Draw the known wear rate for cast iron and for chromium

(2) From the point where the cast iron rate crosses maximum allowable wear, drop a line to intersect the chromium rate line and read across the economic chromium thickness in millimeters or inches.

SUMMARY

Analysis of the data presented indicates the following conclusions:

1. Under any set of circumstances, regardless of fuel, lubricant or type of service, a chromium plated cylinder will outperform a cast iron cylinder money-wise

TABLE 7—*Combined Savings*

Maintenance and Fuel for Life of 2 Cast Iron Cylinders

	1.1 mm or .043 in.	1.6 mm or .062 in.	2.4 mm or .094 in.
Cost & Maintenance of 2 Cast Iron Cylinders	$5280	$5280	$5280
Cost & Maintenance of Chrome Cylinders	—3450	—4145	—4605
Cost & Maintenance Savings with Chrome Plated Cylinders	1830	1135	675
Fuel Savings	1457	3437	4381
Combined Chrome Cylinder Savings	3287	4572	5056

2. With either diesel or heavy fuels and corrosion inhibiting oils, the monetary return on the chromium plated cylinder will be greater than with ordinary lubricants.
3. The point of maximum savings with the chromium plated cylinder and the required chromium thickness is established by the presently accepted limits of wear to oversize of cast iron cylinders (Figure 16).
4. At this number of hours, even though chromium wear is greatly below presently accepted wear limits, the chromium plated cylinder should be removed from service and replaced with another chromium plated cylinder so that fuel saving earnings can continue at the nearly new engine performance rate (Figure 13).
5. The worn chromium plated cylinder having just worn through its chromium layer should be replated and returned to service.

APPENDIX A

Method of Arriving at Average Wear Rate in Cast Iron after Chromium Has Worn Through Conventional Lubricants

Ships in Reference 5 which have worn through .8 mm (.031 in.) of chromium and an appreciable depth into the cast iron can be selected and the average wear rate of the combination of Porus-Krome and cast iron computed. This is shown in Table 3 and results in an average wear of the combination of .077 mm (.003 in.) per 1000 hours for diesel fuel and .21 mm (.0083 in.) per 1000 hours for heavy fuel. Taking the average wear rates in chromium only from Tables 1 and 2 and assuming .8 mm (.031 in.) of porous chromium on the diameter, the hours of operation on chromium only can be approximated.

The difference between total hours and chromium hours gives the running time on cast iron and the difference between total wear and .8 mm (.031 in.) of chromium gives the wear in cast iron. On this basis the wear rate in cast iron is derived.

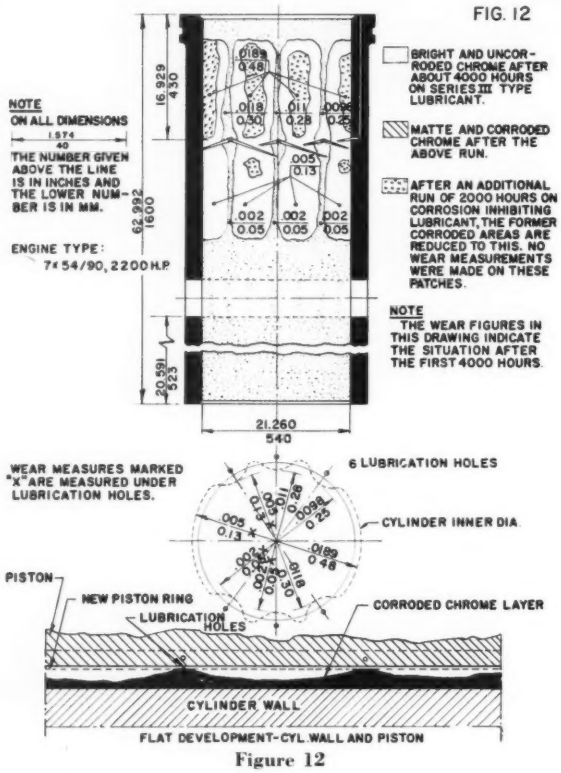
FIG. 12
NOTE
ON ALL DIMENSIONS
1.574
40
THE NUMBER GIVEN ABOVE THE LINE IS IN INCHES AND THE LOWER NUMBER IS IN MM.
ENGINE TYPE:
7×54/90, 2200 H.P.
16.929
430
62.992
1600
20.591
523
21.260
540
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MATTE AND CORRODED CHROME AFTER THE ABOVE RUN.
AFTER AN ADDITIONAL RUN OF 2000 HOURS ON CORROSION INHIBITING LUBRICANT, THE FORMER CORRODED AREAS ARE REDUCED TO THIS. NO WEAR MEASUREMENTS WERE MADE ON THESE PATCHES.
NOTE
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WEAR MEASURES MARKED "X" ARE MEASURED UNDER LUBRICATION HOLES.
6 LUBRICATION HOLES
CYLINDER INNER DIA.
PISTON
NEW PISTON RING
LUBRICATION HOLES
CORRODED CHROME LAYER
CYLINDER WALL
FLAT DEVELOPMENT-CYL. WALL AND PISTON

Figure 12

second cast iron cylinder reaches that on the original chrominum cylinder which occurs around period 8.25-8.75.

Here the chromium plated cylinder begins to make savings again and get back some of its lost profit. In spite of the losing period the plated cylinders all end up with substantial savings. BUT—would a smart accountant or the manager of an investment fund be satisfied with a $1500 (535 14/3) profit (1.1 mm or .043 in. chromium) in 9½ periods when he could have had a profit of $4000 (1428 11/5) in one-half the time? He would not be happy and neither should a ship operator. The right move is simple. The chromium plated cylinder should be removed from service at the point where a cast iron cylinder would have reached its maximum wear limit, i.e., at period 4.75. At this point the plating cost has been earned and the maximum savings have accrued. It should be replaced with another chromium plated cylinder to start earning savings all over again from 0 to 4.75 period on Figure 13. Reference to Figure 7 and 14 will show that the worn chromium plated cylinder has monetary salvage value which may be nearly equal to the price of a new cast iron cylinder. The wear into the cast iron through the chromium is 1.1 mm (.043 in.) for the 1.1 mm (.043 in.) thick chromium; .4 mm (.0157 in.) for the 1.6 mm (.062 in.) thick chromium and 0 for the 2.4 mm (.094 in.) thick chromium.

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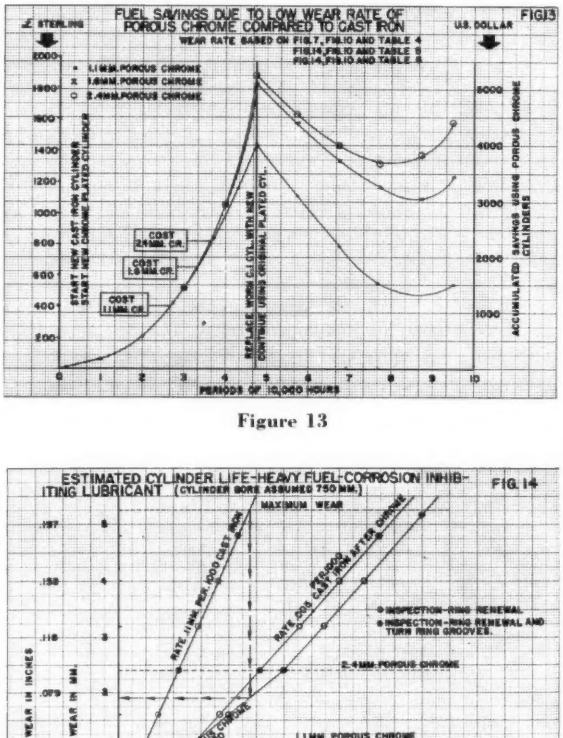
The combined savings for both 1 and 2 cylinder life are plotted on Figure 15. In no case is the combined savings of the 2 cylinder life curve twice as great as the one cylinder life savings, yet it has taken twice as long. It follows that the maximum rate of earning lies in the one cylinder life curve and the optimum economic chromium thickness is


FUEL SAVINGS DUE TO LOW WEAR RATE OF POROUS CHROME COMPARED TO CAST IRON
FIG.13
£ STERLING
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WEAR RATE BASED ON FIG.7, FIG.10 AND TABLE 4
FIG.13, FIG.10 AND TABLE 5
FIG.14, FIG.10 AND TABLE 6
1.1 MM. POROUS CHROME
1.6 MM. POROUS CHROME
2.4 MM. POROUS CHROME
START NEW CAST IRON CYLINDER
START NEW CHROME PLATED CYLINDER
COST 2.4 MM. CR.
COST 1.6 MM. CR.
COST 1.1 MM. CR.
REPLACE WORN CAST CYL. WITH NEW CHROME PLATED CYL.
CONTINUE WITH ORIGINAL PLATED CYL.
ACCUMULATED SAVINGS USING POROUS CHROME CYLINDERS
PERIODS OF 10,000 HOURS

Figure 13

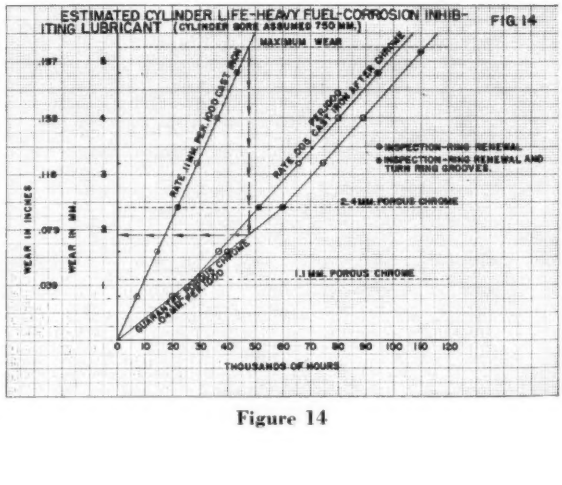
ESTIMATED CYLINDER LIFE-HEAVY FUEL-CORROSION INHIBITING LUBRICANT (CYLINDER BORE ASSUMED 750 MM.)
FIG. 14
MAXIMUM WEAR
RATE 1.1 MM. PER 1000 CAST IRON
RATE .055 CAST IRON AFTER CHROME
INSPECTION-RING RENEWAL
INSPECTION-RING RENEWAL AND TURN RING GROOVES.
2.4 MM. POROUS CHROME
1.1 MM. POROUS CHROME
GUARANTEE PERIOD
WEAR IN INCHES
WEAR IN MM.
THOUSANDS OF HOURS

Figure 14

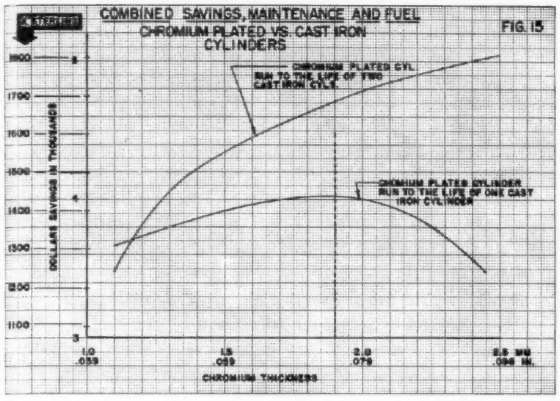
COMBINED SAVINGS, MAINTENANCE AND FUEL
CHROMIUM PLATED VS. CAST IRON CYLINDERS
FIG. 15
CHROMIUM PLATED CYL. RUN TO THE LIFE OF TWO CAST IRON CYLS.
CHROMIUM PLATED CYLINDER RUN TO THE LIFE OF ONE CAST IRON CYLINDER
DOLLARS SAVINGS IN THOUSANDS
CHROMIUM THICKNESS

Figure 15

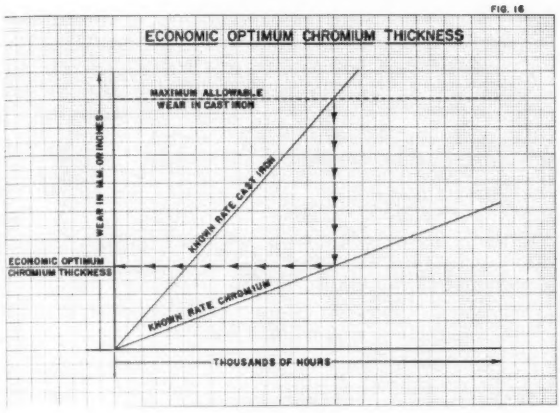

Figure 16

at the peak of the curve for this particular case, at a chromium thickness of 1.90 mm (.074 in.).

As a corollary to this detailed procedure there is a simpler way of establishing the economic chromium thickness for any set of conditions, i.e., any fuel, any type of service or any lubricant, providing the wear rates of cast iron and chromium only are known for that set of conditions. The method follows (see Figure 16):

(1) Draw the known wear rate for cast iron and for chromium

(2) From the point where the cast iron rate crosses maximum allowable wear, drop a line to intersect the chromium rate line and read across the economic chromium thickness in millimeters or inches.

SUMMARY

Analysis of the data presented indicates the following conclusions:

1. Under any set of circumstances, regardless of fuel, lubricant or type of service, a chromium plated cylinder will outperform a cast iron cylinder money-wise

TABLE 7—*Combined Savings*

Maintenance and Fuel for Life of 2 Cast Iron Cylinders

	1.1 mm or .043 in.	1.6 mm or .062 in.	2.4 mm or .094 in.
Cost & Maintenance of 2 Cast Iron Cylinders	$5280	$5280	$5280
Cost & Maintenance of Chrome Cylinders	—3450	—4145	—4605
Cost & Maintenance Savings with Chrome Plated Cylinders	1830	1135	675
Fuel Savings	1457	3437	4381
Combined Chrome Cylinder Savings	3287	4572	5056

2. With either diesel or heavy fuels and corrosion inhibiting oils, the monetary return on the chromium plated cylinder will be greater than with ordinary lubricants.
3. The point of maximum savings with the chromium plated cylinder and the required chromium thickness is established by the presently accepted limits of wear to oversize of cast iron cylinders (Figure 16).
4. At this number of hours, even though chromium wear is greatly below presently accepted wear limits, the chromium plated cylinder should be removed from service and replaced with another chromium plated cylinder so that fuel saving earnings can continue at the nearly new engine performance rate (Figure 13).
5. The worn chromium plated cylinder having just worn through its chromium layer should be replated and returned to service.

APPENDIX A

Method of Arriving at Average Wear Rate in Cast Iron after Chromium Has Worn Through Conventional Lubricants

Ships in Reference 5 which have worn through .8 mm (.031 in.) of chromium and an appreciable depth into the cast iron can be selected and the average wear rate of the combination of Porus-Krome and cast iron computed. This is shown in Table 3 and results in an average wear of the combination of .077 mm (.003 in.) per 1000 hours for diesel fuel and .21 mm (.0083 in.) per 1000 hours for heavy fuel. Taking the average wear rates in chromium only from Tables 1 and 2 and assuming .8 mm (.031 in.) of porous chromium on the diameter, the hours of operation on chromium only can be approximated.

The difference between total hours and chromium hours gives the running time on cast iron and the difference between total wear and .8 mm (.031 in.) of chromium gives the wear in cast iron. On this basis the wear rate in cast iron is derived.

TABLE 8—*Combined Savings Maintenance and Fuel for Life of 1 Cast Iron Cylinder*

	1.1 mm or .043 in.	1.6 mm or .062 in.	2.4 mm or .094 in.
Cost & Maintenance of 1 Cast Iron Cylinder	$2490	$2490	$2490
Cost & Maintenance of Chrome Cylinder	—3025	—3720	—4180
Cost & Maintenance Savings with Chrome Plated Cylinder	— 535	—1230	—1690
Fuel Savings	3993	5158	5173
Combined Chrome Cylinder Savings	3458	3928	3483

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SUMMARY OF WEAR RATES

Material	Rate per 1000 hours				Percent	
	Diesel fuel		Heavy fuel			
	mm	in.	mm	in.	Diesel fuel	Heavy fuel
All cast iron cylinder	.25	.0098	.5*	.019	100.0	100.0
Cast iron after Porus-Krome	.14	.0055	.26	.010	56.0	52.0
Porus-Krome only	.054	.0021	.154	.060	21.6	31.0
.8 mm (.031 in.) Porus-Krome and cast iron	.077	.0030	.21	.0083	31.0	42.0

An approximate check on the wear rates in cast iron after Porus Krome may be made by the following reasoning:

$$\frac{(\text{Rate in cast iron} \times \text{hours in cast iron}) + (\text{Rate in Porus Chrome} \times \text{hours in Porus Krome})}{\text{Total hours}} =$$

Average rate in Porus Krome and Cast Iron together.

Substituting for diesel fuel: $\dfrac{(.14 \times 9.100) + (.054 \times 14.800)}{27.060} = .0755$

This is a fair check on .077 of Table 3.

Substituting for heavy fuel: $\dfrac{(.26 \times 6.300) + (.154 \times 5200)}{11.400} = .214$

which is fair agreement with .26 of Table 3.

* This estimate scaled down from .63 of Table 1.

"THE SHIPBUILDER AND MARINE ENGINE-BUILDER"

CORROSION CONTROL IN TANKERS

ACKNOWLEDGEMENT

This article is an abridged version of a paper presented to The Institute of Marine Engineers on 25 February 1958, as published in the April 1958 edition of "The Shipbuilder and Marine Engine-Builder." The author was A. Logan, Technical Director, Shell Tankers, Ltd.

CORROSION within the cargo and ballast spaces of tankers is probably the major technical problem in pertoleum transport. The extent of the problem has been increased since the war, not only by the great increase in total tonnage, but also by the size of individual tankers from 16,000 tons deadweight to more than 100,000 tons deadweight.

The fleet with which the author is associated numbers more than 200 ocean-going vessels of some 3.4 million tons deadweight. Although accurate figures cannot be given because of the variable nature of the factors concerned, it is estimated that, if no preventive measures were taken, the annual cost of tank corrosion in a fleet of this size could be in the region of £2,000,000.

The purpose of this paper is to outline the corrosion problem and comment on three possible methods of overcoming it. The first part of the paper deals with the cost of corrosion, the pattern of corrosion and the nature of corrosion. In the second part, the author's experience with cathodic protection, paint protection and inhibitor protection are detailed.

THE CORROSION PROBLEM

Cost of Corrosion

The main costs incurred due to corrosion within the cargo and ballast spaces of tankers may be attributed to the following causes:

(a) Structural steelwork renewals.
(b) Renewal of pipe-lines, heating coils, etc.
(c) Loss of deadweight arising from the heavier scantlings adopted to allow for corrosion, and also from scale build-up on structural members.

Demurrage in connection with repair work must be borne in mind, and likewise any losses in earning capacity, which might arise due to difficulties in cleaning or gas-freeing of heavily scaled tanks, or to bulkhead leakages affecting a ship's ability to carry segregated cargoes.

The costs associated with certain of these items are not readily calculable. In the case of structural steelwork renewals, however, actual costs incurred are known and it will be a useful guide to the expenses involved if some of these figures are quoted. Although corrosion data have been collected from the more modern vessels, such as the general purpose tankers of 18,000 tons deadweight, and the larger super-tankers, corrosion costs have not as yet been incurred, and the figures quoted in Table I therefore, are related to older vessels of 12,000 tons deadweight, which have been in alternating white and black-oil service, and to T.2 vessels of 16,000-tons deadweight which have been in predominantly black-oil service. The average age of these vessels is now between 12 and 14 years.

The main periods at which steelwork renewals are made occur at the four-yearly special surveys, and the associated costs, therefore, are cyclic.

In the case of the black-oil ships shown in Table I, steel weights have not been given, since the nature of the repairs in these ships is such that steel weight bears no distinct relationship to costs. It will

TABLE I

Ship	Second special survey				Third special survey			
	Years on:		Total cost of steelwork repairs/ renewals, £	Weight of steel renewed, tons	Further years on		Total cost of steelwork repairs/ renewals, £	Weight of steel renewed, tons
	White oils	Black oils			White oils	Black oils		
12,000 d.w.t.								
Norrisia	8½	*Nil*	200,000	870	2½	2	8,000	Negligible
Northia	8½	*Nil*	216,000	940	4	*Nil*	16,000	50
Narica	7	1½	150,000	600	½	4	93,000	320
Nassarius	6	2	25,000	100	½	3½	280,000	1,080
Neverita	6	2	—	Negligible	*Nil*	4½	146,000	520
Naticina	4	5	69,000	290	1	4	225,000	800
Nassa	3	6½	—	Negligible	2	2	98,000	270
16,000 d.w.t.	Trading practically exclusively on black oils. Wastage of underdeck and associated structure and heavy pitting on bottom and other horizontal surfaces.							
Thallepus							45,000	Cost for fourth special survey is expected to be between £80,000/ £100,000.
Thaumastus							36,000	
Theliconus							48,000	
Tagelus							33,000	
Tectus							26,000	

be noted, however, that repair costs are considerably lower than in the white-oil trade and are incurred later in the lives of the ships.

These figures are intended to indicate the magnitude of the sums involved and to highlight the consequences if no steps are taken to curb corrosion.

Corrosion Pattern

It will be appropriate at this stage to distinguish between the two main service conditions and to deal with their effects on the corrosion pattern. In this context, white-oil service is defined as the carriage of light products, such as gasolines, kerosenes, aviation fuels, etc. Black-oil service refers to heavier oils, such as gas oil, fuel oil, crude oil, etc.

(a) White-oil Service

The cargoes carried in ships engaged on the white or clean-oil trade leave little or no oily film behind on the steelwork of the tanks, and consequently there is little impedence to general overall corrosion and wastage. The corrosion is broadly of a uniform pattern, but will be accentuated at points where repeated localized stressing takes place.

The corrosion rates normally found in such ships will depend upon the usual variables associated with the corrosion of steel, the one of greatest importance being the presence or absence of salt-water. During ballast passages, sea-water is, of course, carried in certain tanks, and it is normal experience to find an appreciably higher rate of corrosion in these tanks than in non-ballast tanks. Typical corrosion loss figures which have been evaluated under known conditions are 0.015 in. per year and 0.0065 in. per year in regularly ballasted and non-ballast tanks respectively, both figures relating to one face only of the corroding part. Table II gives details of thickness losses calculated from

TABLE II

	Northia			*Narica*			*Norrisia*		
	Approximately eight years white oil service			Approximately seven years white and 1½ years black oil service			Approximately eight years white oil service		
	Original thickness, in.	Range of loss, in.	Corrosion rate per side per year, in.	Original thickness, in.	Range of loss, in.	Corrosion rate per side per year, in.	Original thickness, in.	Range of loss, in.	Corrosion rate per side per year, in.
Center Tanks									
Deck girder	0.40	0.08–0.22	0.005–0.014	0.40	0.09–0.19	0.006–0.012	0.40	0.10–0.20	0.006–0.0125
Deck longitudinals	0.43	0.087–0.237	0.006–0.015	—	—	—	0.437	0.18–0.39	0.011–0.025
Deck transverses	0.42	0.07–0.24	0.005–0.015	0.42	0.08–0.17	0.005–0.011	—	—	—
Transverse bulkhead stringers	0.40	0.025–0.175	0.002–0.011	0.40	0.09–0.21	0.006–0.013	All reported thin and fractured.		
Wing Tanks									
Deck longitudinals	0.43	0.02–0.10	0.002–0.006	—	—	—	0.437	0.06–0.11	0.004–0.007
Deck transverses	0.42	0.02–0.10	0.002–0.006	—	—	—	—	—	—
Stringers	0.40/0.42/ 0.44	0.07–0.26	0.005–0.016	0.40/0.42/ 0.44	0.06–0.22	0.005–0.014	0.40/0.42/ 0.44	0.10–0.22	0.006–0.014
Tie beams or struts	0.42	—	—	0.42	0.08–0.20	0.005–0.0125	0.42	0.07–0.22	0.005–0.014
Shell frames	0.44	0.07–0.26	0.005–0.016	0.437	0.010–0.19	0.006–0.012	0.437	0.06–0.21	0.004–0.013
Bulkhead stiffeners	0.44	0.07–0.22	0.005–0.014	0.437	0.010–0.19	0.006–0.012	0.437	0.06–0.21	0.004–0.013

measurements in ships after long periods in the white-oil service, and it will be seen that there is considerable variation in corrosion loss.

When considering the so-called pattern of corrosion, the picture will clearly be incomplete without reference to the various parts of the tank structure and the respective wastages which determine whether or not such parts require renewal within a ship's life. From such a picture it should then be possible to see which are the most vulnerable parts and later to consider means for their protection, either together with the remainder of the tank structure, as is normally done, or, if feasible and economic, perhaps selectively.

A survey of vessels which have spent a high proportion of time in the white-oil service, without any form of protection, has revealed the following pattern:

Center Tanks

(i.) Deck Structure.—Deck girder, deck transverses and deck longitudinals have frequently to be renewed at the second survey, with wastage concentrated particularly in way of the hatch openings.

(ii.) Bulkheads.—Severe wastage is frequently found on bulkhead stiffeners and in way of the stiffener notches in transverse bulkhead stringers. On longitudinal bulkhead plating "grooving" also occurs just above the heavier baseplate. This wastage is caused by severe local corrosion at a point of repeated high stress, and fatigue fractures often result.

(iii.) Bottom Structure.—General overall corrosion is usually less severe on bottom plating, bottom transverses and shell longitudinals. Pronounced localized attack, however, is often encountered on the various horizontal surfaces, particularly in tanks used for ballast purposes.

Wing Tanks

The stringers are normally the first members to require renewal, with general thinning also occurring in shell frames.

Accepting this pattern and using the average corrosion rates previously quoted, it is possible to estimate the useful life of the various parts of the tank structure and this is set down in Table III for a typical modern general purpose tanker of 18,000 tons deadweight. It should be mentioned, of course, that in applying this information, local phenomena will almost certainly distort the picture, while any period in the black-oil trade will tend to increase the life of structural parts.

(b) Black-oil Service

As implied in the foregoing, corrosion in the form of general wastage is the exception rather than the rule in ships engaged in the carriage of black oils. Wastage is, however, frequently found in the ullage space above the normal cargo level and, as would be expected, in any tanks which may be required rarely for cargo purposes.

So far as the bulk of the steelwork in the tanks of black-oil ships is concerned, it has been the author's experience that the residual oily, and in the case of certain crudes, the waxy film which is left adhering to the tank structure provides a reasonably effective barrier against corrosion, provided it is not disturbed or penetrated. Unfortunately, it is virtually impossible to avoid such disturbances, in view of operations such as tank washing, and the exposure of local areas by one means or another can and does often give rise to the very severe local corrosion called pitting in certain of the ship's tanks. This localized attack or pitting is the predominant source of trouble in black-oil ships.

As might be imagined, the most important factor is the presence or absence of sea-water. Pitting trouble is always the most severe in those tanks which are used for ballast purposes, and the relatively slight attack, which is sometimes found in non-ballast tanks, can be attributed almost entirely to the tank washing water lying around on bottom plating and other horizontal parts. It is also noticed, in the case of ballast tanks, that the worst cases of corrosion are found in those tanks which are most frequently used for "clean" ballast water following tank washing; in other words, where protection by oily or waxy films is the least effective.

METHODS OF CORROSION CONTROL

Several fundamental methods exist for overcoming any corrosion problem and many of these have been subject to practical trials for reducing the corrosion in ships' tanks. These methods may be described briefly as follows:

(1) Structural design, *e.g.*, to reduce the hard spots which may cause local high corrosion rates, ultimately leading to fracture.

(2) Use of structural materials of greater corrosion resistance.

(3) Elimination of the main factors on which corrosion is dependent, *viz.*—

(a) Oxygen elimination by blanketing with inert and neutralized flue gases.

(b) Dehumidification or elimination of water by drying the air which enters the tanks to below the critical moisture content.

(c) In this category, also, can be included the use of cold instead of hot salt-water and fresh instead of salt-water for tank cleaning.

(4) Cathodic protection, whereby the steel in a ballast tank is made one large cathode, instead of local anodes and cathodes, by coupling it to a base metal, *e.g.*, magnesium alloy, which is anodic to steel. The anode corrodes and is sacrified to preserve the steel.

(5) Coatings of various types, *e.g.*, the barrier type, which are effective in so far as they prevent

TABLE III—*Anticipated Corrosion in 18,000-ton Deadweight Tankers in White-oil Service. Center Tanks Used for Ballast; Wing Tanks not used for Ballast.*

(1) Structure	(2) Thickness/ ruling section, in.	(3) Thickness at 25 per cent loss unless otherwise stated, in.	(4) Corroding faces	(5) Calculated thickness after: 8 yr., in.	(6) 12 yr., in.	(7) 16 yr., in.	(8) Remarks
Wing Tanks							
Deck plating	0.94	0.705 0.799 (15%)	One side	0.888	0.862	0.838	Above limit.
Shell plating—side	0.99 (sheer) 0.70	0.841 (15%) 0.525	One side	0.648	0.622	0.596	Above limit.
Shell plating—bottom	0.84	0.630 0.756 (10%)	One side	0.788	0.762	0.736	Above limit.
Longitudinal bulkhead plating (upper section), vertical	0.48	0.360	One side	0.428	0.402	0.376	Satisfactory if center tank side protected.
Longitudinal bulkhead plating (lower section), horizontal	0.54	0.405	One side	0.488	0.462	0.436	Satisfactory if center tank side protected.
Transverse bulkhead plating (upper section)	0.49	0.368	(i) Two sides (ii) One side	0.386 0.438	0.334 0.412	0.282 0.386	Below limit after twelve yr. Satisfactory if one side protected.
Transverse bulkhead plating (lower section)	0.52	0.390	(i) Two sides (ii) One side	0.416 0.468	0.364 0.442	0.312 0.416	Below limit after twelve yr. Satisfactory if one side protected.
Deck transverses	0.43	0.323	Two sides	0.326	0.274	0.222	Near limit after eight yr.
Deck longitudinals	0.43	0.323	Two sides	0.326	0.274	0.222	Near limit after eight yr.
Horizontal girders (upper, middle and lower)	0.43	0.323	(i) Two sides (ii) One side	0.326 0.378	0.274 0.352	0.222 0.326	Near limit after eight yr. Near limit after sixteen yr.
Shell frames and top brackets	0.43	0.323	Two sides	0.326	0.274	0.222	Near limit after eight yr.
Bulkhead stiffeners and end brackets	0.43	0.323	Two sides	0.326	0.274	0.222	Near limit after eight yr.
Bilge brackets	0.43	0.323	Two sides	0.326	0.274	0.222	Near limit after eight yr.
Bottom transverses	0.43	0.323	Two sides	0.326	0.274	0.222	Near limit after eight yr.
Bottom longitudinals	0.51/0.68	0.383	Two sides	0.406	0.354	0.306	Below limit after twelve yr.
Center Tanks							
Deck plating	0.94	0.705 0.799 (15%)	One side	0.820 0.820	0.760 0.760	0.700 0.700	Border line after sixteen yr. Below limit after twelve yr.
Shell plating (bottom)	0.84	0.630 0.756 (10%)	One side	0.720 0.720	0.660 0.660	0.600 0.600	Below limit after sixteen yr. Below limit after eight yr.
Longitudinal bulkhead plating (upper section), vertical	0.48	0.360	One side	0.360	0.300	0.240	Limit reached in eight yr.
Longitudinal bulkhead plating (lower section), horizontal	0.54	0.405	One side	0.420	0.360	0.300	Below limit after twelve yr.
Transverse bulkhead plating (upper section)	0.50	0.375	(i) Two sides (ii) One side	0.260 0.380	0.140 0.320	0.020 0.260	Below limit after eight yr. Border line after eight yr.
Transverse bulkhead plating (lower section)	0.53	0.397	(i) Two sides (ii) One side	0.290 0.410	0.170 0.350	0.050 0.290	Below limit after eight yr. Below limit after twelve yr.
Deck center girder	0.43	0.323	Two sides	0.190	0.070	—	Below limit after eight yr.
Deck center girder face flat	0.80	0.600	Two sides	0.560	0.440	0.320	Below limit after eight yr.
Center line vertical web	0.48	0.360	Two sides	0.240	0.120	—	Below limit after eight yr.
Deck transverses	0.43	0.323	Two sides	0.190	0.070	—	Below limit after eight yr.
Deck longitudinals	0.43	0.323	Two sides	0.190	0.070	—	Below limit after eight yr.
Bottom center girder	0.50	0.375	Two sides	0.260	0.140	0.020	Below limit after eight yr.

TABLE III (continued)

(1) Structure	(2) Thickness/ ruling section, in.	(3) Thickness at 25 per cent loss unless otherwise stated, in.	(4) Corroding faces	(5) Calculated thickness after: 8 yr., in.	(6) 12 yr., in.	(7) 16 yr., in.	(8) Remarks
Bottom center girder face flat	1.27	0.953	Two sides	1.030	0.910	0.790	Below limit after twelve yr.
Bottom transverses	0.46	0.345	Two sides	0.220	0.100	—	Below limit after eight yr.
Horizontal girders (stringers)	0.44	0.330	(i) Two sides (ii) One side	0.200 0.320	0.080 0.260	— —	Below limit after eight yr. Below limit after eight yr.
Shell bottom longitudinals	0.51/0.68	0.383	Two sides	0.270	0.150	0.030	Below limit after eight yr.
Gusset plates	0.62	0.465	Two sides	0.380	0.260	0.140	Below limit after eight yr.

NOTES: 1: The above figures are given principally to show the relative vulnerability of the various structural members if corroding at the rates referred to in the text.
2: It is not intended that the figures in column 3 be interpreted as Classification requirements.

access of oxygen and water to steel surfaces, the combined barrier/inhibitor type or the metallic type which cathodically protects steel surfaces.

(6) Corrosion inhibitors which may be:

(a) Oil soluble and added to the cargo.

(b) Water soluble and added to the ballast water.

(c) Oil or water soluble and sprayed on to the walls of empty tanks.

Some success has been achieved in designing to eliminate hard spots. Although experiments with corrosion-resistant materials have been carried out, they are not used except for special items such as heating coils, and mild steel is still the traditional construction material. In practice, therefore, the problem is one of controlling the corrosion of mild steel by one or more of the basic methods (3) to (6) listed in the foregoing.

CATHODIC PROTECTION

About five years ago, an investigation of cathodic protection was carried out in a white-oil ship of 12,000 tons deadweight, and the encouraging results of this work were given in a previous paper. Subsequent tests in a large crude-oil carrier, in which class of ship the problem of localized corrosion, or pitting, was becoming of much concern, indicated that this method of protection was feasible under black-oil conditions also. Although further developments of some aspects were obviously necessary, and performance data at that time scarce, the decision was made to apply cathodic protection to the ballast tanks of all new construction and a number of existing ships.

Approximately 135 ships in the author's company now have their ballast tanks protected in this way and, apart from certain exceptions where conditions have been unfavorable, observations made at annual dockings have indicated that a high level of protection is being achieved.

The chief features which make cathodic protection attractive as an anti-corrosion measure are:

(a) Installation of anodes is straightforward and involves little delay or interference with other work.

(b) Application can be made with almost equal facility in new construction and in older ships.

(c) Existing rust and scale is detached.

The principles of cathodic protection have been well-described and depend briefly on the difference in electrical potential between the steelwork and the protecting anode. As seen in the case of the millscale/steel couple, there exists a sizeable difference in potential, sufficient for an appreciable current to flow from the steel to the millscale when there is a water path to complete the electrical circuit. In a similar, but much more pronounced way, the difference in electrical potential which exists between steel and a metal such as magnesium results in a current flow when the two are coupled together in sea-water. In this case, however, the magnesium becomes the anode and corrodes while the steel (and the millscale, if any) becomes the cathode and is protected: that is, provided sufficient current is generated in this way while the structure is submerged in water.

It is generally accepted that full protection is not reached the precise moment a tank is filled with water ballast, but is attained over a short period of time, depending on the anode current available and on a number of other factors. The success of any cathodic protection system, therefore, will depend partly upon the rapidity with which a steel structure, such as a cargo tank, receives the full benefit of the protective current from the anodes each time it is ballasted.

One of the first reactions to take place in a cathodically protected tank is the removal of existing rust and scale. The quantity of scale detached will clearly depend upon the condition of the tanks involved, varying from very little in the case of new ships to several tons per tank in old white-oil ships.

Detachment will generally be completed during the first month or so in service under average ballast conditions, but this period may be as long as six months, or even longer, in the case of ships which have been in white-oil service for some years prior to protection.

A further reaction takes place during and immediately following descaling. This is the deposition of some of the calcium and magnesium compounds present in the sea-water ballast. This calcareous deposit, which is of a highly alkaline nature, is usually taken as a visual indication of the effectiveness of a cathodic installation. Ideally, the film should be fairly thin, hard and adherent, but very little work has been done so far with a view to controlling its physical nature.

In addition to the obvious benefit of the anode currents themselves in suppressing corrosion during the carriage of sea-water ballast, the role played, or believed to be played, by the calcareous film should be noted. Firstly, there are several advantages of such a coating:

(a) Provided it is undisturbed, it will give a fair measure of protection after the tank has been deballasted. Observations indicate that, under optimum conditions in a dry empty tank, a high degree of protection may in fact be given over several months. On the other hand, if less favorable conditions obtain, rusting will take place behind the film, causing its eventual detachment.

(b) Once formed, it will assist in the rapid attainment of full protection during subsequent ballastings, of obvious value where ballast times are relatively short.

(c) It will cut down the current taken from the anodes during subsequent ballastings, resulting in lower anode consumption and improved economy.

On the other hand, a calcareous deposit may prove to be a disadvantage if, for example:

(a) It is laid down as a soft voluminous coating under conditions which are yet to be satisfactorily explained, offering little protective "blanketing" and possibly adversely affecting gas-freeing.

(b) The physical conditions in the tank after deballasting are such that the coating is soon detached, in which case corrosion can proceed apace, the scale eventually in turn to be dislodged and added to the debris to be lifted.

Some damage to or disturbance of a calcareous film must, of course, inevitably take place from time to time, for example, during tank washing, and it will obviously be beneficial to reduce such operations to the minimum consistent with operation requirements. It should, at the same time, be pointed out that such damage or detachment will normally be made good by further deposition during a subsequent ballasting or, on the other hand, if unfavorable conditions have obtained in the meantime and rusting has taken place behind the film, a new basic film will be formed on the metal surfaces.

An appreciable quantity of hydrogen gas is evolved when a protected compartment is ballasted, and precautions must be taken to see that there is an escape route for this gas to the atmosphere.

TANK COATINGS

Coatings, often of a bituminous nature, have been used successfully for many years for the protection of peak tanks, etc., where the alternating conditions are water-filled or empty. There is increasing scope for the use of this type of coating, under similar conditions, in the permanent ballast tanks which are a feature of super-tankers.

For a coating which is to be used in cargo tanks, chemical resistance to all types of petroleum hydrocarbons and to water is a prerequisite. The coating must not be harmed by the cargo, nor be harmful to the cargo, and it must protect the steel surfaces against corrosion whether tanks are ballasted, in cargo or empty. In addition, and most important, the coating must lend itself to application under normal shipyard conditions. In recent years a number of chemically resistant paints, *e.g.*, vinyls, isocyanates, epoxies, etc., have been developed, which appear to be capable of fulfilling these requirements.

The author's experience has been mainly with epoxy coatings, which first became available in Europe in 1953. This type of coating is cured by the hardener, which is added to the paint before application. The curing depends on the temperature, but is not dependent on the presence of oxygen.

Service trials with epoxy coatings were started in 1954, the coatings being applied to the surfaces of constructed tanks prepared *in situ* by different methods. In 1956, a second series of trials was initiated to explore the possibilities of devising a simplified and cheaper coating procedure for new ships, based on the preparation of the steel before assembly.

Service Trials with Epoxy Coatings

Experience with epoxy coatings applied after *in situ* preparation of tank surfaces is summarized in Table IV.

Different methods of surface preparation, *i.e.*, shot-blasting, flame-cleaning, and chipping and wire-brushing were examined to establish the type of preparation required to give the necessary long coating life. Tanks were coated both in summer and winter months to ascertain that the epoxy coatings can be satisfactorily applied in all seasons. The test in the *Omala* was arranged to investigate the suitability of the coating process for old, white-oil ships. The suitability of epoxy coatings for use in conjunction with cathodic protection was investigated, since protection of the upper parts of ballast tanks by the latter method alone presents certain problems. Paints were brush-applied, except for the last two coats, with one of the four proprietary paints used in the *Hemiglypta*.

TABLE IV—*Summary of Tests with Epoxy Coatings*

Ship	Coating applied	Area	Surface preparation	Number of coats and average coating thickness	Curing	Service conditions	Condition of coated area
Koratia	May 1954	Upper and lower parts of 1 center tank.	Shot blasting Small area chipped and wire-brushed.	3 3-4 thou. in.	Natural.	Black and white oils/ballast.	Good condition after 3 years. Slight local rusting and blistering noted after 1 year, but has not increased significantly. Extensive film breaking due to mill scale detachment on wire-brushed area after 1 year.
Korovina	Aug. 1954	1 center tank.	Flame cleaning.	4 4-5 thou. in.	Natural.	Black and white oils/ballast.	After 1 year coating completely intact. No pinpoint rusting. Loss of adhesion on approximately 10-15 per cent of surface after 2 years, becoming progressively worse in third year.
Shell Welder	Feb. 1955	2 complete tanks.	Shot blasting.	3 3-4 thou. in.	Accelerated with hot air.	White oil.	Good condition after approximately 2½ years. Local rusting and blistering after 6 months, but not increased significantly since.
Hyala	Mar. 1955	Deckhead area in 2 wing tanks and 2 center tanks. Cathodic protection also fitted in one of center tanks.	Shot blasting.	3 3-4 thou. in.	Accelerated using infra-red dull emitters. Ambient temperatures about freezing.	Wing tanks: black and white oils. Center tanks: black and white oils/ballast.	Good condition after 2 years. Local rusting and blistering after 1 year which varied in extent among 4 proprietary paints used. No apparent effect of cathodic protection on the paint films.
Hemiglypta	Mar. 1955	Deckhead area in 2 center tanks. Cathodic protection also fitted in one.	Shot blasting.	3 3-4 thou. in.	—ditto—	Black oil/ballast.	Good condition after 2 years. Slight local rusting and blistering. General corrosion on mechanically damaged area. No apparent effect of cathodic protection on the paint film.
Omala (16-year old white oil ship)	Aug. 1955	Upper part of one center tank.	Shot blasting.	4 4-5 thou. in.	Natural.	White oil/ballast.	Good condition after 1 year.
Khasiella	Jan. 1956	1 center tank.	Shot blasting.	4 4-5 thou. in.	Accelerated with infra-red dull emitters. Ship afloat, air and water temperatures very low.	White oil/ballast.	After 1 year fairly widespread rust spotting, slight local blistering but no breakdown of film.
Hima	Aug. 1956	6 wing tanks.	Sand blasting.	4 4-6 thou. in.	Natural.	White oil.	At only inspection after 5 months, coatings in very good condition.
Kosicia	Sept. 1956	2 wing tanks. 1 center tank.	Shot blasting.	4 4-6 thou. in.	Natural.	White oil.	No inspection at time of writing.

Coatings can be expected to facilitate tank-cleaning operations, both by the prevention of corrosion and by presenting a smooth non-absorbent surface. Some practical information on this point was obtained in the *Korovina*, where it was reported that cleaning of black oil from a coated tank was completed in half the time for a non-coated tank.

In the *Shell Welder*, two further tanks were spray-coated with aluminum. Apart from slight local detachment, where mill scale removal was incomplete, the coatings remained in good condition after two years' white-oil service.

Performance of Coatings

Good surface preparation is of paramount importance if epoxy coatings are to give effective long-term protection. On chipped and wire-brushed surfaces, extensive detachment of the coating due to poor adhesion occurred in the first year. On flame-cleaned surfaces, the coating remained perfect for one year, but, during the second year, corrosion under the coating caused breakdown. Neither of these two methods is capable of entirely removing mill scale, which is essential, and the only satisfactory method for preparing surfaces for epoxy coatings is blasting with abrasives.

Epoxy coatings on shot-blasted surfaces show no significant breakdown after three years' service and nothing to suggest that failure is impending. There is no evidence of chemical attack on the coatings, even by heated cargoes, and no wear has occurred, the average film thickness remaining unchanged. High-pressure jets of hot water from tank-cleaning machines appear to have no effect on the coatings.

The main defects have been local rusting and

blistering in the film. Examination of different systems, applied and used under identical conditions, shows that the extent of blistering is variable, suggesting that the phenomenon is not inherently connected with epoxy coatings.

Coatings applied in winter months, when ambient temperatures were very low and heating was required to provide satisfactory curing, are in good condition after two years' service. If provision is made, therefore, for heating when temperatures are low, epoxy coatings can be applied satisfactorily under all normal conditions.

The effective life, *i.e.*, before major breakdown occurs, of epoxy coatings applied over shot-blasted surfaces, cannot yet be predicted. It may be concluded, however, that the life will be, at least for black-oil ships, considerably in excess of three years.

Coating Process

Before operations start, all construction and installation work in the tanks should be completed, with the exception of the installation of heating coils. The heating coils may suffer damage, particularly if of light gauge, and will impede effective cleaning and coating of the tank bottoms. Heating-coil bearers should, however, be installed before coating and all steel retaining clips, fittings, etc., associated with the coils should be coated before installation. Coating of fastenings, etc., is particularly necessary if copper-based alloy coils are involved, since uncoated steel fittings would form small anodic areas and suffer accelerated corrosion.

Surface Preparation

The surfaces should be blasted to a whitemetal finish, and this requires large volumes of compressed air at adequate pressure if it is to be achieved consistently. Experience indicates that, if a suitable pressure is to be maintained, independence from normal shipyard air supplies may be advisable.

The first coat of paint should be applied to the prepared surfaces before re-rusting occurs.

Painting

Ideally, the surfaces and atmosphere should be clean and dry. Vacuum cleaning is the only satisfactory method of removing dust and debris before painting. It is recommended that the first coat, at least, should be applied by brush, since this method is preferable when surfaces are dusty. Spray application of subsequent coats appears to be satisfactory, and, from the practical point of view, has the advantage of being quicker. Below about 55° F, curing of the paint is slow, and heating preferably with infra-red dull emitters, will be required. In these circumstances, heat can also be used to dry the surfaces and the atmosphere before painting. The finished coating should always be left to cure for at least seven days before the tank is put into service.

DEVELOPMENT OF A SIMPLIFIED PROCEDURE FOR COATING CARGO TANKS IN NEW SHIPS

The United States Navy has recently reported encouraging data on the acid descaling of tanks in an old ship; the spent acid was dumped at sea.

For new ships, mechanized shot-blasting of the steel before fabrication and assembly offers an alternative approach. Suitable equipment is available and, if suitably fitted into production lines, steel plates and sections can be mechanically shot-blasted at relatively low cost. The question of installing suitable facilities has been discussed with shipbuilders, who have explored the possibilities individually. As a result, and despite initial obstacles, several mechanical units for shot-blasting both steel plates and sections will come into operation in this country within the next few months.

Surface preparation before fabrication raises a new problem, *viz.*, that of protecting the steel before final painting. To obtain maximum benefit from the shot-blasting, the prepared steel should be immediately protected against corrosion.

Some possible types of protectives have been examined, but all have serious practical limitations. Preservative oils are unlikely to give long-term protection and would, in any case, require removal before painting. Sprayed metal, *e.g.*, aluminum coatings, cause welding problems. Paints have been found, in practice, to suffer extensive mechanical damage during fabrication, which may not always be immediately evident, and, in general, they must be removed from areas which are to be welded. Production requirements would permit application of one coat only of a very quick-drying paint, which normally would give a film thickness of approximately 0.001 in. Cast round steel shot (30 mesh) is being recommended for use in mechanical units, and with this grade of abrasive, a film thickness of approximately 0.0025 in., requiring probably three normal coats of paint, would be required.

The technical advantage of preventing re-rusting of the shot-blasted steel cannot be over-emphasized and the search for an acceptable shop coating continues. If this cannot be found, the only alternative is to leave the prepared steel unprotected until after fabrication and assembly and then re-treat it before painting. Hand sand-blasting will be necessary for cleaning welds, but is not acceptable for general re-treatment. In practice, it has been found that shot-blasting to remove rust from plates which are already free from mill scale is just as slow as the cleaning of raw plates. Other methods of re-treating the rusty, but mill scale-free, plates after fabrication and assembly are being examined in service trials. These alternative methods include wire-brushing alone or after flame heating to dehydrate the rust. Treatment of wire-brushed surfaces with phosphoric acid is also being examined.

CORROSION INHIBITORS

Although it can be done successfully, the application of epoxy coatings to tanks in old, white-oil ships is not regarded as attractive owing to the slowness and difficulty of the operation. Cathodic protection which can be readily installed in any tank is limited in use to ballast compartments. There remains, therefore, a requirement for a convenient method of protecting the non-ballast tanks in old, white-oil ships. The use of inhibitors may well provide a solution to this problem, while at the same time being a possible alternative to painting for new, white-oil ships.

Oil soluble inhibitors, which are either added to the cargo or sprayed on to the walls of empty tanks, may be used. Alternatively, water soluble inhibitors sprayed on to the walls of empty tanks may be employed and this is the method preferred.

Selection of Inhibitor

An immediate choice might be the alkaline sodium nitrite inhibitor, which is used successfully in the United States of America for the prevention of corrosion in white-oil ships. It was established, however, that this inhibitor adversely affects the quality of certain products; in particular, it increases the water-haze retention in aviation turbine fuels. Alternative inhibitors were examined, therefore, in a laboratory corrosion test, and, from the results, the most promising was selected for shipboard trials. The selected inhibitor was one of the most effective of the compounds tested and, furthermore, has no adverse effect on the range of oil products.

Practical conditions were simulated as far as possible in the laboratory experiments. Since, in actual practice, the steel surfaces in cargo compartments will always be, to some extent, rusty when the inhibitor is applied, previously rusted steel panels were subjected to a cycle lasting 28 days, including spraying with sea-water, spraying with inhibitor, storage in a moist atmosphere in the presence of sea water, and a short exposure by immersion in moist white-oil products. This cycle was repeated, so that the complete tests lasted 56 days. After the tests, the steel specimens were de-rusted and the corrosion rate, expressed in thousandths of an inch per year, was calculated from the weight loss.

The influence of the concentration of the selected inhibitor on the corrosion rate of steel is shown in Table V. It will be noted that the highest level of protection was obtained at a concentration of about 15 percent.

TABLE V—*Influence of Inhibitor Concentration on Corrosion Rate (Thousandths of an Inch per Year)*

Inhibitor concentration per cent	Exposure time	
	0-28 days (1 cycle)	0-56 days (2 cycles)
0	5.8	6.7
5	7.0	5.6
10	4.0	3.5
15	2.1	2.0
30	2.5	3.8

Method of Application

Parallel to the inhibitor test program, experiments were carried out with various types of equipment for spraying and atomization. The distribution of the inhibitor is of prime importance, and the aim is to obtain at least 80/90 percent coverage of the steel with an adequate amount of inhibitor.

Two methods of spraying a solution on to tank steelwork have been used in tankers in the U.S.A. The first of these involves a fixed Christmas-tree type of installation of spray nozzles in each tank, located so that all the steelwork will be wetted by direct action. Inevitably, particularly to give complete coverage of a wing-tank structure, a large number of sprays are required, and the installation is costly to erect and maintain.

The second method involves a rotating spray arm in each tank. The arm rotates in a horizontal plane while, simultaneously, spray heads, mounted at each end of the arm, are rotated in a diagonal plane.

This method is more attractive than the fixed sprays, but it still involves a costly installation and, in so far as both methods rely on directional control, they will be prone to a fall-off in efficiency in the absence of regular maintenance and checking.

The shortcomings of these spray systems have given rise to the development of a method of introducing the inhibitor in the form of a very fine fog, the principle being that the liquid in suspension will migrate uniformly to the steelwork without any external mechanical agency. Distribution is effected by means of natural and convective air currents, and the critical factor is that the fog should be sufficiently fine to persist long enough for it to be carried on to all surfaces in the tank.

The first efforts in this direction involved the use of a mechanically-driven rotating disc-type fog-maker of the design used widely in horticulture. With this machine, a coverage of approximately 50 percent of the surface was achieved, but there was excessive fall-out of large particles to the tank bottom. The fog was not persistent and underside surfaces were consequently neglected.

It was clear that a finer particle size was necessary and the steam atomizer principle was considered most likely to give this. In addition, it was thought that the increased temperature from the use of steam would assist in convective circulation. Tests, therefore, were made with a steam atomizer of the type used in oil-fired boilers and, again, a coverage of about 50 percent was achieved, although fog persistence was improved.

It now appeared that increased turbulence was necessary and that more than one atomizer would be required per tank. The final tests, therefore, were made using a steam atomizer, which creates its own turbulence by means of entrained air. Indications

TABLE VI—*Approximate Costs of Cathodic Protection and Painting of Tank Interiors (inclusive of Installation Costs, etc.)*

	18,000-d.w.t. ship		31/32,500-d.w.t. ships		
	11 center tanks	22 wing tanks	8 wing tanks (ballasts only)	12 wing tanks (cargo/ballast)	2 center tanks (cargo/ballast)
Areas, sq. ft.	171,000	279,000	126,000	191,700	59,760
Cathodic protection for "back oil" service	£	£	£	£	£
Initial	7,000	12,000	—	8,000	2,000
Recurring 4 yearly	5,200	9,000	—	6,000	1,500
Total for life of (a) 8 yr.	12,200	21,000	—	14,000	3,500
(b) 12 yr.	17,400	30,000	—	20,000	5,000
(c) 16 yr.	22,600	39,000	—	26,000	6,500
(d) 20 yr.	27,800	48,000	—	32,000	8,000
Cathodic protection for "white oil" or "ballast only" service					
Initial	11,000	19,000	12,000	—	—
Recurring 4 yearly	8,200	12,500	9,000	—	—
Total for life of (a) 8 yr.	19,200	31,500	21,000	—	—
(b) 12 yr.	27,400	44,000	30,000	—	—
(c) 16 yr.	35,600	56,500	39,000	—	—
(d) 20 yr.	43,800	69,000	48,000	—	—
Painting costs (epoxy paints)—in situ *surface preparation*					
Initial	25,600	41,900	18,900	28,800	9,000
Recurring costs unknown, but total costs would be:					
(a) Assuming one "equivalent repainting" for life of 20 yr.	51,300	83,600	37,800	57,500	18,000
(b) Assuming two "equivalent repaintings" for life of 20 yr.	77,000	125,500	56,600	86,400	27,000
Painting costs (epoxy paints)—prior surface preparation					
Initial	19,000	31,000	14,000	21,300	6,600
Recurring costs unknown, but total costs would be:					
(a) Assuming one "equivalent repainting" for life of 20 yr.	38,000	62,000	28,000	42,600	13,200
(b) Assuming two "equivalent repaintings" for life of 20 yr.	57,000	93,000	42,000	63,900	19,800

NOTES: (i) Figures quoted include known or anticipated yard costs, profits, etc.
(ii) The cost of *in situ* surface preparation and painting has been taken as 3s. per sq. ft.
(iii) Prior surface preparation and *in situ* painting costs have been taken as 2s. 3d. per sq. ft.
(iv) Paint coatings are assumed to be spray-applied to a thickness of approximately 0.005 in.

are that three of these fog-makers installed in the wing tank of an 18,000-ton tanker will give a coverage figure of at least 95 percent, using about 40 gallons of inhibitor solution per tank.

CONCLUSIONS

Of the methods described, cathodic protection is limited to tanks used for ballast purposes. It is subject to operational procedure, but may be applied with almost equal facility both to new construction and existing ships. An initial installation presents little difficulty and usually no delay to a ship is involved. Renewals of anodes at intervals of several years likewise present no major problems.

Painting may be used for any cargo or ballast condition, and has now reached the stage in its development when it is beginning to be seriously considered for new construction either for whole tanks or for selected parts, provided interference with work normally proceeding during construction and any delay to completion are both of an acceptable order. On the other hand, the problems inherent in the descaling and coating of tanks are of such magnitude that, for existing ships, painting is at present wholly unattractive. As might be imagined, the high initial cost of painting tank interiors has been a major deterrent to the use of this method. That the painting of new ships is now becoming more attractive is partly due to the long lives expected of some of the present-day coating materials and partly due to the appreciable savings which may be effected by mechanized surface preparation prior to fabrication —a matter which is being actively pursued by shipbuilders in this country and abroad.

An obvious role for a paint coating is the protection of "non-ballast" tanks, but the fact must not be overlooked that the scope for painting extends in principle to "ballast" tanks also. It may even be advantageous, in certain circumstances, for both cathodic protection and a paint coating to be used simultaneously in a ballast tank. The situation might also arise, for example, where it has been economic to paint a ballast tank initially, but where it proves impracticable to do any major repainting after some years have elapsed. In such a case, cathodic protection could be provided for the remaining life of the ship.

Regarding the third method dealt with, *viz.*, inhibitor application, it is believed that the inhibitors so far considered will find application only in the "non-ballast" spaces of white-oil ships. This is, of course, the main outstanding problem to be solved and the inhibitor method may prove to be suitable and economically attractive, not only for "non-ballast" tanks in new construction, but also for similar tanks in existing ships. The effectiveness of the method, of course, has still to be determined over a period of time, and it is not yet possible to say whether the method is likely to become a serious competitor to painting for those conditions where it is applicable.

Before a decision can be reached on a method or methods to be used, not only must the technical and practical merits, as already outlined, be considered, but an economic evaluation must also be made. In this connection, approximate overall costs are summarized in Table VI, in so far as they relate to the two methods, cathodic protection and painting, for which figures are available.

It is intended that the data given in Table VI serve as a guide only, as the variations which are to be expected between jobs of this sort preclude the giving of exact figures.

In the case of the larger ships built essentially for the black-oil trade, only those center tanks likely to be used for ballast purposes have been included in Table VI, since it is not thought that the cost of "complete" protective measures in the remaining center tanks would be justified. On the other hand, for such "cargo only" tanks, local protection may be provided by painting, say, deck members or bottom plating, if experience suggests that particular trading conditions may give rise to serious trouble.

Receiving investigation in the search for high energy aircraft fuels are pyrophoric substances. These are liquids which ignite spontaneously in air. Pyrophoric fuels are particularly adapted to ramjets, since the device can be made smaller and simpler than one designed for more conventional fuels. The lighter weight, in turn, requires less fuel for the same performance, so that the savings accumulate. As an example, it is estimated that a Mach 1.5 ramjet designed for a target drone can be produced for $600. With hydrocarbon fuel, the unit would have to be scaled up to include flame-holders, ignition system, and larger fuel tanks at an estimated cost of $5,000 to $10,000. Pyrophoric fuels also have promise as mixing agents in hydrocarbon fuels to improve rocket performance. The three most prominent members of the pyrophoric family are TEA (tri-ethyl aluminum), TEB (tri-ethyl borane), and TMA (tri-methyl aluminum).

—from MECHANICAL ENGINEERING
September 1958

Official U. S. Navy Photograph

Artist's conception of the world's first nuclear aircraft carrier, USS ENTERPRISE, now under construction at Newport News Shipbuilding and Dry Dock Company. The use of nuclear fuel, providing the ability to make long runs at sustained high speeds, adds a new order of magnitude to the mobility of this versatile weapons system.

"THE WELDING JOURNAL"

TODAY'S TREND IN SHIP-HULL RESEARCH

ACKNOWLEDGEMENT

This article was published in the June 1958 Welding Research Supplement of "The Welding Journal." The author is David K. Felbeck, who is associated with the National Academy of Sciences—National Research Council, Washington, D. C.

INTRODUCTION

No longer can ship research produce an order-of-magnitude reduction in ship failures such as resulted from the studies during and just after World War II. No longer can a ship be considered satisfactory if it can just keep from falling apart, although keeping a ship in one piece was a serious problem not too many years ago (Figure 1). The early empirical studies effectively revealed those improvements in design, materials and fabrication necessary to reduce to a very low level the catastrophic failure of welded merchant ships and tankers. Thus problems of size, structural loads, seaworthiness and availability of quality materials have displaced brittle fracture as the major applied problems today. Although further refinements have resulted from more recent applied studies, no extensive contributions toward improvement of ship hulls can be expected except from careful and thorough fundamental investigation of several pertinent fields. Thus an examination may be in order of significant results from the current program of the only organization in the United States devoted exclusively to ship-hull research.

Figure 1. Forward half of the Russian-operated American-built Liberty ship S. S. Valeri Chkalov, which broke in two in the North Pacific on Dec. 13, 1943.

The Ship Structure Committee was formed to pool the efforts of four agencies (Coast Guard, Navy, Maritime Administration and American Bureau of Shipping) in ship-hull research. It is assisted in planning and administering a research program by two committees of the National Academy of Sciences-National Research Council: the Committee on Ship Structural Design and the Committee on Ship Steel. With the further help of liaison representatives from organizations such as the Welding Research Council, the Ship Structure Committee research program represents the combined efforts of most United States groups concerned with ship problems. An interpretation of the current trend of this research program may serve to emphasize the importance of fundamental studies to a field as applied as ships. The results of this fundamental work must, of course, be further developed before they can influence ship construction; hence, an appreciable portion of the Ship Structure Committee program has been devoted to applied studies that owe their origins to earlier fundamental results from this or other programs. The relative growth of fundamental research within the Ship Structure Committee program since its formation in 1946 is shown in Figure 2.

BRITTLE-FRACTURE STUDIES

Studies on initiation of brittle fracture have shown that prior cold working of steel plates in tension perpendicular to the direction of the test load can cause a reduction in test load at fracture to about 70 percent of the load originally required to produce plastic deformation (yield load.) If the prior cold working is in compression, the fracture load may be reduced to 59 percent of yield load. It is possible that these results may help to explain fracture of ships at loads thought to be safely below the yield load. Investigators concerned with fundamentals of crack propagation have shown conclusively that a dynamic strain wave of a magnitude several times the yield strain precedes a rapidly propagating crack. Another investigator is studying independently the relationship of local strain rate to the conditions that are associated with fracture. These projects should help to explain part of the sensitivity of brittle steels to running cracks.

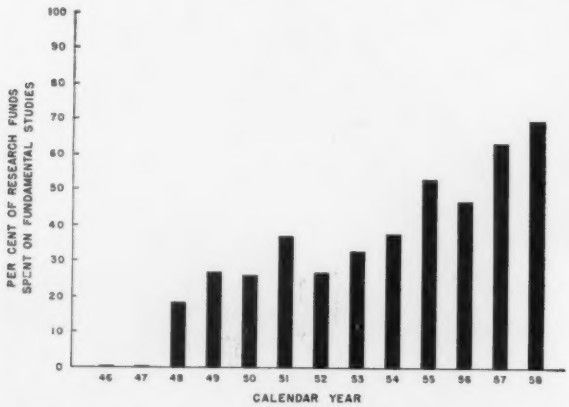


Figure 2. Percent of Ship Structure Committee research funds spent for fundamental research during each year since 1946.

Three projects that are under way or will begin shortly are part of a world-wide study to develop a reliable test for brittleness in steel. A small project nearing completion was set up to find whether there is any correlation between the variation with temperature of the appearance of a laboratory-produced fracture surface on a specimen of ship plate and the temperature at which the ship fractured in service. Because of contradictory reports of the influence of manganese in reducing brittleness in steel, a careful series of experiments is being planned to isolate as well as possible the effect of varying manganese content and to study the influence of this variation on different kinds of tests. The third project is an applied study to assist Commission IX of the International Institute of Welding, which is concerned with brittle fracture, in its current comparison of fracture-appearance tests with energy-absorption tests. This study will be eflected through two-way exchange and testing of steel samples of known pedigree.

A comparison of riveted and welded crack arresters has verified that, for equal materials, the riveted arrester is better (because of the discontinuity of the plating), but that flush-welded strips of high notch-tough steel can stop running cracks. The required width of material and properties of the arrester strip have not yet been determined.

An investigation of the influence of metallurgical structure on fracture has obtained additional results on three grades of steel that support an earlier conclusion that the ferrite grain size is important but does not fully explain the brittle behavior of steel. A detailed study of pre-yield phenomena at low temperatures has shown that the generally accepted dislocation pile-up theory of cleavage fracture does not explain, by a factor of more than 1000, the local strain prior to fracture. Furthermore, careful examination of both flat and cylindrical specimens has revealed numerous microcracks formed just prior to fracture at low temperatures. Under certain conditions, several of these microcracks can join up to form a single larger crack.

Ship Structure Committee experience of recent years has demonstrated that the most productive brittle-fracture studies have been those that most closely investigated fundamental mechanisms through simplified tests at very low temperatures, at very slow loading speeds or involving specimens of the simplest type. Except for accidental revelation of new insight, the present high degree of un-

derstanding of *what* happens prior to and during fracture, as obtained through extensive and large-scale testing, cannot be expected to add appreciably to the understanding of *why* a material behaves as it does.

HOW IMPORTANT ARE FLAWS?

All known ship fractures, and most nonship brittle fractures of structures, originated at a flaw, void or structural discontinuity. The severity of structural discontinuity can be controlled by proper design. The size, severity and frequency of flaws can be partially controlled by proper procedures in the steel mill and during welding. But until the influence of a flaw on fracture initiation is better understood, it will not be possible to specify precisely how large and severe a flaw must be to warrant replacement of that portion of the structure.

Although quantitative evaluation of flaws is yet to be achieved, one important experimental observation has emerged from studies of flaw behavior; the magnitude and pattern of residual stresses in the region of a flaw, as well as the shape and size of the flaw, can profoundly affect the performance of a structure. The combined influences of better understanding of fracture mechanics and improved residual-stress measurement should bring closer the goal of precise evaluation of flaw severity. In this field there is clear need for fundamental studies rather than direct empirical work.

Flaw-detection methods are better developed today than are the flaw-evaluation criteria. A combination of fundamental and empirical studies has produced heartening progress over the last few years. The flaw-detection research program has been comprehensive in its studies of radiological methods. Tests have been completed for a handbook on the use of available gamma-ray sources for ship-hull inspection. New methods are being studied for detection of flaws through use of a quantitative radiation counter instead of the usual photographic film. Recent work indicates it may be possible, through X-ray scattering techniques, to inspect ship hulls with all men and equipment on only one side of the plate. A complete survey of the ASTM standard weld-flaw specimens by ultrasonic means is currently under way as part of a program to investigate all possible methods for improving the speed and accuracy of flaw detection. Other recently developed techniques—graphic and numerical—are planned for later inclusion in this program.

MAKING BETTER STEEL

One of the most effective early solutions to the brittle-fracture problem in ships was the specification of certain grades of steel by the classification societies. Together with improvements in design and fabrication, the use of better grades of steel has reduced the ship-casualty rate to almost zero. Although the ship-fracture problem has thus been brought under control for the present, one critical new problem and one important region for improvement in steelmaking still demand attention from research groups. The new problem is to improve notch toughness of steel of the increased thickness that results from design of today's much larger merchant vessels, particularly tankers. The needed improvement in steelmaking is the replacement of fully killed steel with a cheaper steel requiring less elaborate manufacturing facilities. The current efforts of the materials-improvement phase of the Ship Structure Committee research program are directed toward practical solutions to these two questions.

Fundamental studies of the brittle-fracture mechanism that should provide greater confidence in the properties of thicker steels have been described above. Pending such better understanding, shipbuilders will continue to avoid very thick plating in large ships through use of double plating or additional internal structural support. Because these expedients present additional fabrication problems and are usually more expensive, the Committee on Ship Steel has recommended a study to determine what is the best possible killed steel for thick plate, including composition, practice and heat-treating variables. An improved killed steel may be able to fulfill temporarily the requirements for thick steel of high quality until the fracture behavior of thick plate is better understood and the most suitable material can be selected.

In an effort to produce experimental semikilled steels that would be acceptable alternatives to the fully killed steels now in use in thicknesses over one inch, trial heats were made that proved to be as good as the present steels in thicknesses to 1¾ in. (Figure 3). Preliminary results of tests in progress on three full-scale production heats of this high-manganese low-carbon semikilled steel appear to verify the conclusions from the experimental heats.

The growing use of controlled-temperature mill rolling of steel plate in a number of countries abroad has been followed with considerable interest by investigators in this country, even though it is doubtful whether American mills could at present tolerate the reduction in capacity that would result from controlled cooling of slabs during the rolling process. Several controlled-rolled plates have been obtained from one European steel producer by the Ship Structure Committee for study. A careful examination of the gross and fine structure of these plates may reveal, in conjunction with other parallel studies in brittle fracture, some of the factors that determine the reported improvement in quality. If the improvement from controlled rolling can be satisfactorily related to service behavior, this mill practice will provide manufacturers and users with

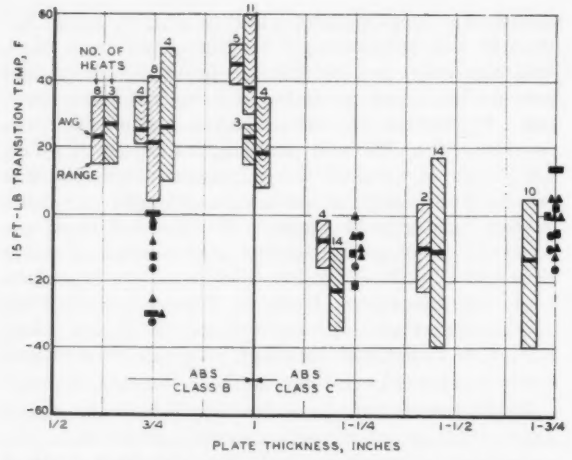


Figure 3. V-Notch Charpy 15 ft-lb transition temperatures of experimental heats of high-manganese low-carbon semi-killed steel (shown as individual points of various shapes, as reported by several laboratories) as compared with American Bureau of Shipping classes of steels in the same thickness ranges (shown cross-hatched). (Vanderbeck, SSC-101)

an additional alternative to fully killed steel for some applications.

The quality of currently produced steel must be known accurately in order to evaluate the effectiveness of any proposed change in steelmaking practice. One project, conducted jointly with the American Iron and Steel Institute, has developed simplified mill sampling methods based on a thorough study of a small number of heats of semikilled and killed ship steel from several commercial mills. Tentative conclusions indicate that two or three samples from different locations within a single ingot can provide a measure of the average values and range of values for an entire heat. This sampling procedure is now being used on a number of commercial heats to determine the range of properties within a heat and among several heats. Another testing program has studied steel samples obtained directly from shipyards to find out whether current steel going into ships is better than the old wartime steel. No recent data have contradicted earlier conclusions that current steel used in ship construction is markedly better, with an average transition temperature 43° F below the average of samples taken from wartime fractured ships.

DESIGNING BETTER SHIPS

Within recent years improved understanding of the nature of ocean waves has provided hope for ultimate fulfillment of the designer's dream of knowing all loads acting on a ship. If development of knowledge of the load system were accompanied by corresponding refinements in design procedures, the stress pattern throughout the ship could be calculated. Then the ship designer could utilize materials of known properties with much greater confidence and thus could surely improve the design of a ship and effect savings in weight of structural material. The final product of this long-term research program would be a safer ship, cheaper to build and capable of carrying a greater payload.

Research in several fields will be able to contribute to these improvements in ship design. Until recently the fields of oceanography and structural design, for example, had nothing in common, but soon the work of the investigator studying shape and frequency distribution of waves will vitally affect the design loads used by the ship designer. In an effort to relate such diverse research efforts, the Society of Naval Architects and Marine Engineers and the Ship Structure Committee are jointly sponsoring preparation of a monograph describing the current state of knowledge on waves, ship response to waves and the resultant forces on and stresses in a ship. This report will provide a basis for planning research programs in the fields that most require attention.

A group under the Committee on Ship Structural Design is already preparing recommendations for a comprehensive long-range research program that can be used as a guide for future studies directed toward achieving rational design of ships. In this study the investigators are utilizing the monograph mentioned above and other pertinent literature. A second group is preparing a report of current methods of structural analysis and design in other fields that may be useful in ship design.

These planning projects will provide the foundation for a long-term research program that will require the joint efforts of many groups. Much of the work of the first few years will be fundamental in nature to develop sound bases for understanding and predicting phenomena such as ocean waves, thermal stresses in structures and the influence of inelastic deformations. These studies will not all bear fruit immediately in the form of improved ship design. Many of the research results will contribute only indirectly by providing facts upon which further research of a more applied nature will depend. Out of the latter work should come, in stages, the ship hull improvements that are the goal of the combined research programs.

ACKNOWLEDGEMENT

A description of the Ship Structure Committee research program would be incomplete without at least a list of the research organizations and people who have been or are actually performing the work. Those groups connected through recent or current research contracts with the Ship Structure program described herein are as follows:

Batelle Memorial Institute—P. J. Rieppel
Brown University—Prof. D. C. Drucker, Prof. C. Mylonas

University of Illinois—Prof. W. J. Hall, Prof. R. J. Mosberg, Prof. W. H. Munse, Prof. N. M. Newmark
Massachusetts Institute of Technology—Prof. B. L. Averbach, Prof. W. A. Backofen, Prof. Morris Cohen, Dr. W. S. Owen
National Bureau of Standards—J. A. Bennett, C. L. Staugaitis, Dr. M. L. Williams, Dr. W. J. Youden
Naval Ordnance Laboratory—E. L. Criscuolo
Naval Ordnance Test Station—Dr. J. I. Bujes
Naval Research Laboratory—Dr. J. M. Krafft, W. S. Pellini, P. P. Puzak
New York Naval Shipyard—E. A. Imbembo, N. A. Kahn*
New York University—Prof. George Gerard
Southwest Research Institute—Dr. Edward Wenk, Jr.
Stevens Institute of Technology—Prof. B. V. Korvin-Kroukovsky, Prof. E. V. Lewis
Swarthmore College—Prof. S. T. Carpenter, Prof. R. F. Linsenmeyer

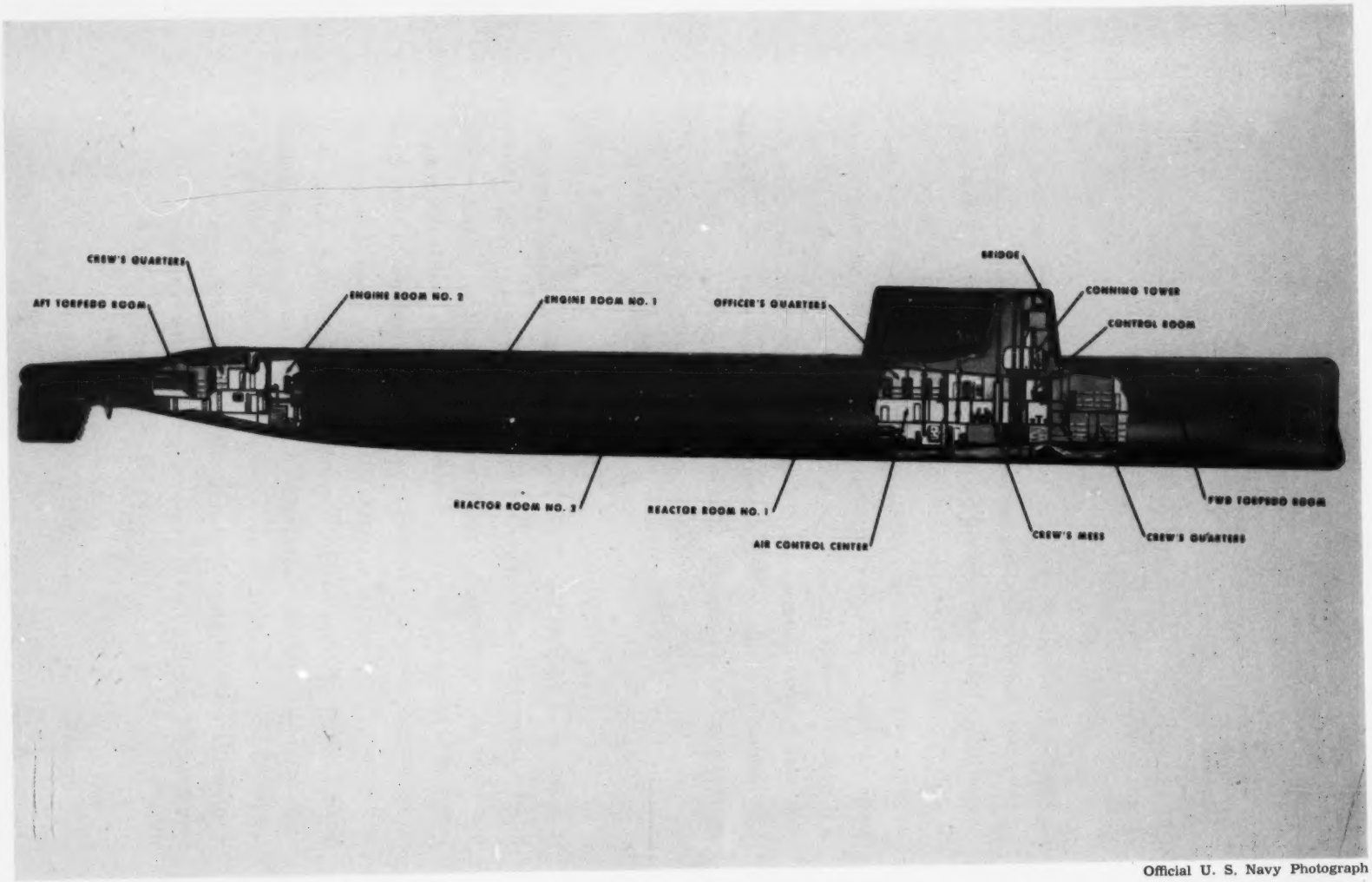
* Deceased.

The Experimental Boiling Water Reactor at the Argonne National Laboratory was shut down in May because of a turbine failure. Investigation revealed that the failure resulted from stress concentration due to a notch in a blade root, causing the blade to become detached from the rotor. No evidence was found of stress corrosion or any effects caused by the steam from the reactor. Of particular interest is the fact that the maintenance was conducted on a routine basis, since the level of radioactivity almost immediately after shutdown was 2 milliroentgens per hour. Current radiation protection standards limit exposure to 3000 milliroentgens in a three month period, or an average of 5000 milliroentgens per year.

—from MECHANICAL ENGINEERING
October, 1958

In studies of the possibilities of a lunar base, Dr. Peter Castruccio of the Westinghouse Electric Corporation has proposed a lunar power station operating on the solar-cell, or photo-electric, principle. Because of the lack of atmosphere on the moon, components normally small in size and encased in the envelope of a vacuum tube can be assembled on a large scale in the open. This proposal envisages the use of large sheets of a coated plastic, covering several acres. The photo-sensitive coating would omit electrons, which would be collected by a wire mesh parallel to and slightly separated from the plastic. Power generation periods would be 14 days in length—the length of a lunar day. It is estimated that the equipment would have a weight of only about three pounds per kilowatt of capacity.

—from MISSILES AND ROCKETS
September 22, 1958



Official U. S. Navy Photograph

USS *Triton*, world's largest submarine, launched at the Electric Boat Division of the General Dynamics Corporation in August, is a symbol of the increasing role of undersea warfare in our national defense.

ROBERT J. STEELMAN

ENGINEERING EVALUATION OF OF SHIPBOARD ELECTRONIC EQUIPMENT

THE AUTHOR

is a B.S. in Engineering Physics, Ohio State University, June 1938. Electronics Engineer at Aircraft Radio Laboratory, Wright Field, Dayton, Ohio, until 1946. Electronics Engineer at US Navy Electronics Laboratory, San Diego, California 1946 to present. Currently Deputy Head, Evaluation Branch, Engineering Division, USNEL. Member Institute of Radio Engineers and of Sigma Pi Sigma, national physics honorary society. Registered professional engineer.

About 50,000,000 years ago there was a caveman named Oog. Seeing a pterodactyl munching ferns below his cave, he grabbed his spear and hurried out to replenish the family larder. On the way he passed his friend Guk, who was sunning himself on a rocky ledge.

"Hey!" Guk called. "Where are you going in such a hurry, boy?"

Oog waved his spear. "I haven't got time to talk! See that big critter down there in the swamp? With the price of meat what it is—"

"Relax," Guk said, "and let me take a closer look at that spear." Over Oog's protests he took the spear and examined it critically. "Just as I thought. You have been using some of those crummy vines to bind the head on, instead of rawhide." He twisted the stone point, and it came loose in his hand.

"You broke it!" Oog shouted. "You broke my spear, just when I need it!"

Guk was patient.

"This is all for your good. Vines rot, lying around in a damp cave."

Oog tore a double handful of hair from his chest. "Those were Grade A vines—I picked them myself!" His eyes narrowed. "You *cut* them somehow or other. It's a trick!"

Guk blinked mildly.

"Okay, knucklehead. Go out and get yourself killed. I was only trying to help."

Oog looked doubtful. "Well—"

"Here." Guk fumbled in his bag. "Take some of these thongs and tie the point on *right*. This is prime brontosaurus hide."

"But the pterodactyl! He'll get away!"

Guk sat back down in the sun and folded his arms. "Maybe so, friend, but an empty stomach is better than a hole in your head."

So Oog missed his pterodactyl, and Mrs. Oog (who was a prominent clubwoman) berated him bitterly, and all the little Oogs were hungry and went to bed with rumblings in their stomachs. But other animals lurked in the ferns below, and Oog, with his improved spear, became in time a mighty hunter, although he was not inclined to credit Guk for any part of his success.

The fable has several points. Guk was the first evaluation engineer. One does not have to look for parallels with the modern science of evaluation engineer-

ing, either. Guk's client, Oog, had no time to waste in having anyone test his weapon. He was impatient with the whole process. And when Guk found the weapon to be defective, Oog argued long and loud, even suggesting there was trickery involved. When he finally achieved success with his redesigned weapon, he was sure he had done it all by himself, and had only harsh words for Guk.

To return to modern times, the field of evaluation engineering is of paramount importance to the Navy, and to other services, all of which maintain extensive establishments and facilities devoted to this function. In these times of a growing and complex technology, evaluation has taken on increased importance. The only effective control over such a technology is the process of evaluation, carried out logically and objectively by trained engineers and scientists. For this discussion we are concerned with the following evaluation standards:

a. Written specifications or contract requirements which may have been established for the equipment.
b. Suitability for Naval use.
c. The state of the art, and the general standards of good engineering design and practice.

These three standards may not necessarily agree. Anyone who writes specifications will tell you that no specification ever completely established the performance of an equipment. Moreover, an equipment may fail to meet a specification requirement and still be perfectly acceptable for Naval use; all this means is that the specification was wrong. And an equipment may represent fine engineering design, carried out well in practice, and yet not be suitable for Naval use. It is apparent, therefore, that the process of evaluation can be a risky business, beset with booby-traps, and requiring a scientific approach.

The evaluation process is necessary in many fields. Ordnance, electronic gear, hoists and winches and boilers, aircraft instruments—all these and more are grist for the evaluation mill. They come to the evaluation engineer for a final decision. The finality of his decision is the key to the importance of the process. The evaluation process represents the *last effective*

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Figure 1. Navy Electronics Laboratory, San Diego, California.

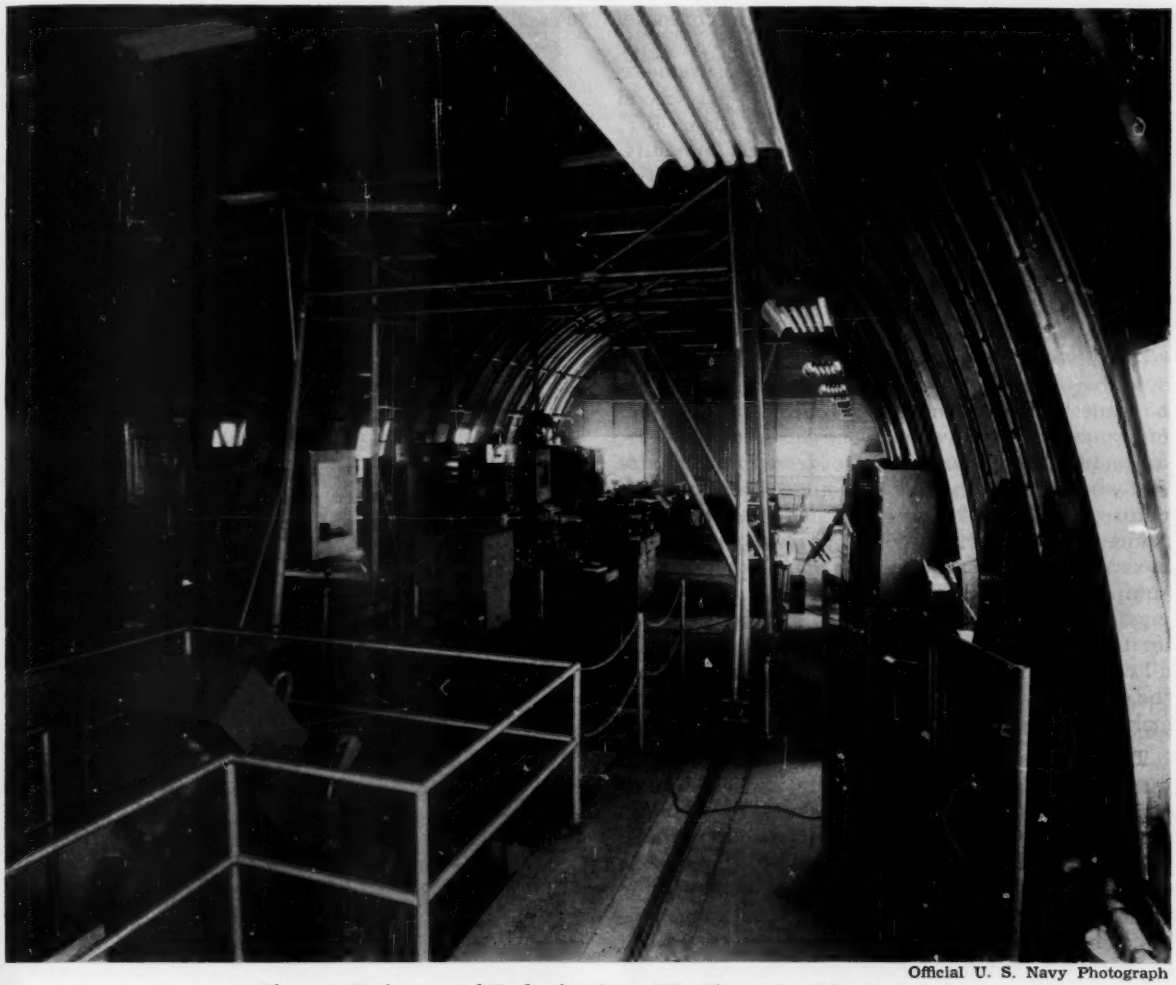

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Figure 2. Environmental Evaluation Area, USN Electronics Laboratory.

viewpoint from which an equipment, or a device, or a material, can be observed prior to release for production and consequent widespread distribution to the military.

There is absolutely no substitute for an unbiased and objective evaluation organization. With the best conscience in the world, a contractor for military equipment may tell you that his equipment will function with complete success down to a hundred degrees below zero centigrade. Perhaps he thought it did, but often it does not. A military organization may have a pressing need for a new miniature radio receiver, and our defense may be hampered without it. But someone has to argue with the military, and point out that this particular item of equipment is unreliable. A few good jolts and bangs in combat will put it out of service, with even more dire results (since tactics may have been based on its use) than if the military had never had it to begin with. Conversely, there is no way to get this necessary objectivity except by establishing an impartial and competent agency which cannot be subjected to pressure by either the seller or the buyer of the equipment. This is an axiom.

For over ten years the Evaluation Branch of the Navy Electronics Laboratory has been in existence as such an agency. Hundreds of electrical and electronic equipments have been evaluated here, and hundreds of reports written. Although the Branch contains other sections, the evaluation effort is confined largely to two; the Electronic Evaluation Section, and the Mechanical Evaluation Section, which operate in conjunction to perform the general engineering evaluation of electronic equipment for the Bureau of Ships. Facilities and instrumentation of the Branch are extensive, and constitute a showplace for frequent visitors to the Laboratory. One of the world's large low frequency vibration machines with a table eight feet across, and capable of 50,000 pounds force; a high-impact shock machine with a hammer

which weighs three thousand pounds; hundreds of thousands of dollars worth of the latest electronic instrumentation, permitting accurate measurements from fractional cycles to kilomegacycles per second in the diversified fields of radio, radar, sonar, IFF, and electronics in general; temperature, humidity, and altitude chambers, including one costing two hundred thousand dollars, with interior dimensions of eight by eight by sixteen feet. As a conservative estimate, the capital facilities of NEL which are engaged in this field are worth over a million dollars. Is it worthwhile? Does it pay its way? What are the findings of the NEL operation that justify a huge and continuing investment by the Navy?

Perhaps the best justification of the program can be made by the statement that, out of the hundreds of equipments involved, *not one has undergone the evaluation process without the detection of faults.* Many of these faults have, of course, been minor in nature, but many of them have been of major significance, often requiring a complete redesign in one or more respects. In some cases, the manufacturer of the equipment has been unable to remedy the shortcomings, and the design specialists of the Navy Electronics Laboratory have accomplished the redesign. In all cases, the report of the Laboratory has indicated the direction and extent of the necessary changes to make the equipment suitable.

The results of this comprehensive evaluation process have been analyzed statistically, and yield interesting results.* The results are discussed in detail in what follows for two reasons; first, the kinds of faults and their statistical distribution will be of interest to designers and Naval technical people as indicating where the common pitfalls are; second, some pertinent comments can be made with respect to each category of failure to establish guide lines which may help prevent repetition of such failures.

The most frequent fault observed in these equipments submitted for evaluation was that of *inadequate electrical design.* This category includes those electrical design shortcomings which are detectable which introduces excessive noise, or peak power in equipment, under normal ambient conditions of pressure, temperature, and humidity. Typical examples might be poor RF amplifier design in a radio receiver which introduces excessive noise, or peak power in a radar set which is below the required level. There is not a great deal which can be said about this kind of fault, except that it should not have occurred. In practically every case, it was possible to effect a remedy by relatively simple means. It is of interest to note that this kind of shortcoming almost never involved a performance requirement which was impossible to meet. This point seems to indicate that writers of specifications are not requiring the very best performance of which the electronic industry is capable. Perhaps there should exist a greater proportion of failures caused by more stringent specification requirements, if only to insure that electronic designers are working closer to creative saturation. Too many operational requirements and written specifications seem to be evolved by asking only that the performance of the AN/ZZZ equipment be 10 percent better in all respects than the previous AN/YYY equipment.

The next category of failures is that traceable to *shock and vibration,* which are here dealt with together because their effects are interrelated, and the mechanism producing the failures has at its roots the same basic physics: the phenomenon of resonance. Most people are familiar with resonance in one form or another. Timed pushes to a child's swing keep it going at a rate dependent on the length of the rope. A tuned electrical circuit oscillates at its resonant frequency when energy is coupled to it to replace natural losses. In the same way, the various parts which make up an electronic equipment vibrate when mechanical energy is coupled to them, and their vibration is extreme if energy components in the exciting waveform coincide in frequency with the natural resonant frequencies of the parts.

So far as shipboard equipment is concerned, the range of frequencies of interest extends up to 33 cycles per second. Mechanical energy from screws and shafting is coupled to electronic equipment via hull plates, deck plates, beams, girders, and intermediate structures. For example, a four-bladed propeller rotating at a maximum speed of 375 RPM will have a frequency of 4 x 375 cycles per minute, or 25 cps. For other applications (aircraft, missiles, etc.) the frequencies of interest may be much higher, reaching several thousand cps. The basic mechanism of shock and vibration damage is the same, however. Exciting energy is coupled to the structure under question, and *wherever a mechanical resonance exists within the equipment, mechanical energy will be absorbed and the resonant part or structure will be excited.*

Simple parallels serve to illustrate the effect. The relationship for resonance in a simple series electrical circuit is

$$f = \frac{1}{2\pi\sqrt{LC}} \text{ cps} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

where L is the series inductance and C is the series capacity. At a frequency determined by the values of L and C, the circuit is resonant, and a maximum of power is absorbed by the circuit. In the same way, the resonant mechanical frequency of a simple beam bearing a weight at one end and clamped in a fixed support is

$$f = \frac{1}{2\pi}\sqrt{\frac{3EIg}{WL^3}} \text{ cps} \quad \ldots\ldots\ldots\ldots\ldots (2)$$

where W is the magnitude of the weight, L is the distance in feet between the weight and the point of

* NEL Report No. 581, "A Statistical Analysis of Electronic Equipment Failures in Evaluation" by R. J. Steelman, 21 February 1955.

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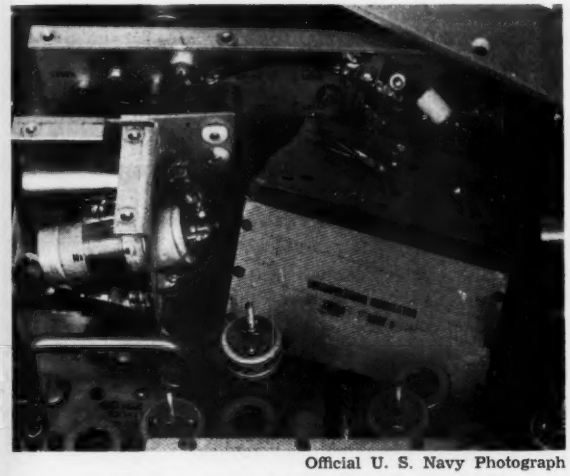
Figure 3. 50,000 Pound Force Reaction Vibration Machine.



Official U. S. Navy Photograph

Figure 4. Typical Vibration Damage.

clamping of the beam, E is the modulus of elasticity of the beam, I is the moment of inertia of the beam, and g is the acceleration of gravity. In general, the resonant frequency of a simple mechanical system with one degree of freedom is

$$f = \frac{1}{2\pi}\sqrt{\frac{\text{spring constant}}{\text{mass vibrated}}} \quad \ldots\ldots\ldots\ldots (3)$$

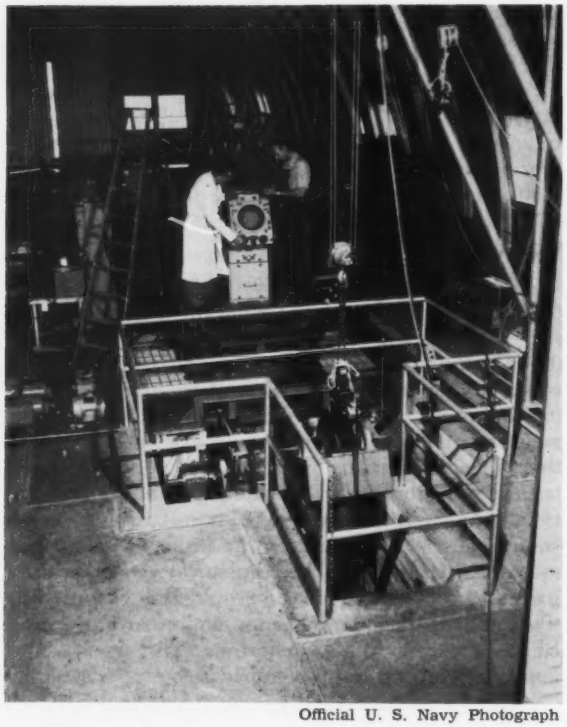
The parameters is these expressions need not be discussed in detail here; the intent is to point out a similarity of the expressions for mechanical and electrical resonance.

Unless the actual phenomenon of resonance is to be made use of, it is usual in electrical design to keep resonant frequencies well removed from the operating frequencies. In the same way, a mechanical system must be protected from mechanical resonances if its performance in vibration is important. Vibration mounts are often employed for this purpose. But vibration mounts which are sufficiently soft to isolate an equipment over the range of shipboard vibration frequencies are usually too soft to protect against shock blows (unless a large excursion is provided for). A preferable method is to use methods of fabrication and mounting which *raise the resonant frequency of all parts above the range of shipboard vibration frequencies.* If parts are so designed and so restrained in their assembly, none of them can absorb energy from the driving source, and the effects of vibration are minimized.

A simple illustration shows the effect. A power transformer may be mounted by the base on a chassis, constituting a cantilevered beam. But suppose the mounting area of the base is small, and the center of gravity of the transformer relatively high above the mounting base. This combination can result in a mechanical resonant frequency of only a few cps for the transformer, causing it to tear loose under sustained vibration (with resulting secondary damage to other parts). On the other hand, if the top of the transformer is restrained with a brace or clamp, the resonant frequency of the transformer can be raised well above 33 cps, with the result that vibration in the shipboard frequency range no longer has appreciable effect on the transformer. The transformer now constitutes a beam clamped at both ends.

Heavy vacuum tubes are often mounted only by the base, leaving the envelope free to vibrate. Tube retainers, while they serve the purpose of keeping the tube in the socket, do little to restrain motion at the top of the tube. A large transmitting tube can have a resonant frequency well within the shipboard vibration range, and moderate vibration can destroy it completely. But a suitable clamp which restrains movement of the envelope can raise the resonant frequency of the whole assembly to the point where it is no longer susceptible to shipboard vibration effects. The same clamp can also restrain excessive motion under shock, so that the envelope does not fracture from collision with nearby components.

Anything which will raise the resonant frequency of a part or assembly or equipment above the range of frequencies to be encountered (or move it below, which is usually less practicable) is likely to be of value in improving the vibration performance. Changing cantilevered beams to beams fixed at both ends has been given as an example. In addition, the maximum attention must be given to *stiffness* and *lightness* in mechanical design. In general, light parts and structures have high resonant frequencies. Stiff assemblies have correspondingly high natural frequencies. Mass distribution within an assembly subjected to vibration is also important. The movement of a mass about an axis of vibration should be kept as small as possible, which is to say that the heavy components in a chassis should be kept as close to the plane of the mounts as possible. All of these cautions, however, are only another way of saying *raise the resonant frequency of all parts above the range*



Official U. S. Navy Photograph

Figure 5. Sonar Set Being Prepared for High-Impact Shock Tests.

Official U. S. Navy Photograph

Figure 6. Typical Shock Damage.

of shipboard vibration frequencies. Only in this way can the assembly, and the parts within, be decoupled from harmful vibration forces.

Shock is a related matter. In general, requirements for a specified performance in the presence of shock are intended to protect the equipment against the effect of near-misses by underwater mines and torpedoes. Investigations have shown that near-misses can raise an entire vessel of many thousand tons several inches in the water, and the resultant shock impulse is impressive in its ability to damage all kinds of equipment.

While the mechanism of a shock blow is slightly different, the end results are similar to those resulting from vibration. To draw again on an electrical analogy, the case is similar to excitation of a resonant circuit by a single sharp electrical pulse. The exciting pulse can be analyzed by the Fourier process to show that it is composed of a large number of harmonically-related sinusoids. These sinusoids form, in effect, an envelope of energy which can extend (depending on the steepness of the wavefront of the pulse) over a great range of frequencies. As a result, almost every single part of an electronic equipment subjected to such a shock blow can find somewhere in the pulse envelope a frequency component of the right range to excite it. In the final analysis, the effects of such shock blows reduce to vibration problems, and similar remedial measures are indicated.

In either case, the rule of thumb is this: *design so that all frequencies will be above the range of frequencies to which the equipment is to be subjected.* Since the structure between a shock machine (or ship) and the resonant part or assembly acts as a low-filter, the higher-frequency components of the shock pulse are attenuated, and do not reach the part or assembly in their original amplitude. Therefore, parts or assemblies having high resonant frequencies are not so likely to be affected by the shock pulse. Also, failure is usually caused (except in brittle materials) by yielding, which is directly related to displacement (such as vibration at large amplitudes). Since parts or assemblies with high resonant frequencies deflect less during excitation, they are less likely to be stressed beyond their yield point.

In practice, the effects of vibration appear in four principal areas: amplification of vibration through the use of resilient mounts and through structural inadequacy, chassis motion due to lack of sufficient restraint (slides, guides pins, etc.), and various miscellaneous faults (chafed cables, etc.). These areas are dealt with in detail in the following paragraphs.

Amplification of vibration through the use of resilient mounts, such as vibration mounts, is easy to avoid by not using such mounts. It has been fully demonstrated that good mechanical design, with adequate margins of safety, can produce electronic equipment to withstand shipboard shock and vibration requirements *without* the use of such mounts.

Vibration mounts, particularly, should never be used with shipboard electronic equipment, since they will probably amplify the intensity of shock blows.

Amplification of vibration through structural inadequacy is usually a simple matter of *lack of stiffness of the enclosure*. The designer should use structural members with high moments of inertia and minimum weight. Adequate cross-bracing should be used in the three principal axes. The mounting of the structure should be such as to prevent excessive racking or rocking within the test range of vibration frequencies.

Excessive chassis motion during vibration ("hammering") is usually the result of lack of guide pins to restrain the chassis in the closed position. A good rule of thumb is this: drawers or other removable chassis should be provided with at least two guide pins on the rear of the chassis.

Poor performance in shock is generally manifested by broken vacuum tubes, broken rivets and screws, cracked welds and castings, deformation of shelves and brackets, drawers rolling out, protective covers popping off, and heavy components tearing loose. Design for protection against shock is a relatively new field, and one which is often left to a designer unfamiliar with Naval requirements. Mere provision of shock mounts is usually not enough; incorrect mounts, or improperly located and/or installed mounts, can actually intensify shock damage. One of the most common sources of trouble in shock is that which comes about due to the use of inadequate bend radii in sheet metal. Stress concentrations resulting from such small bend radii may eventually cause failure even though surface cracking of the bend is not immediately apparent.

Inadequate strength or thickness of material often causes trouble in shock or vibration; nevertheless, the remedy is usually *not* a heavier material, but rather the use of die-formed ridges for stiffening, reinforcement by braces or auxiliary welded members, and similar techniques. Mere "beefing-up" of plates and girders and similar supporting structures usually leads to trouble. The equipment that is larger and heavier than need be is usually the one that gives trouble in shock and vibration. As a spectacular example, one contractor built a device that was found by the Laboratory to be six inches taller, eight inches deeper, and six inches wider than was necessary. Magnetic amplifiers were mounted on one-quarter inch thick steel plates (at ten pounds per square foot). Three unnecessary heavy steel channels were welded to the enclosure for mounting, where four holes through the back of the cabinet would have sufficed. This "battleship" construction was precisely the wrong approach, and the performance of the equipment under shock and vibration was poor because the heavy structures involved were resonant at a low frequency, lying well within the shipboard range of vibration frequencies. A light stiff structure would have given improved performance.

In summary, the following points can be made with respect to shock and vibration problems in shipboard electronic equipment:

a. The basic design goals for proper performance in shipboard shock and vibration should be *stiffness* and *lightness*, with the intent of raising all resonant frequencies well above 33 cps.

b. Vibration mounts should not be used for electronic equipment intended for shipboard installation. Well-designed equipment does not require such mounts, and the mounts may actually intensify shock damage.

c. Most electronic equipments can be mounted without shock mounts, relying instead on the natural resilience of the structure to cushion the equipment.

The next most important category of failure is surprising in view of the many good references on the subject.* *Poor control design* includes inaccessible controls, controls requiring excessive force to operate, parallax, poor visibility of figures, controls with inadequate range, etc.

Briefly, there are certain "expected relationships" between control and effect which should be provided. For instance, wherever possible control movements should be parallel to the axis of display motion which they effect. Controls should be oriented to fit normal "habit-pattern" reflexes, since an operator will normally move his control to center deviations or reduce errors indicated by the visual display. Control dimensions should take into account the normal hand-grasp limitations; adjustment knobs for electronic equipment, for instance, should be limited to about two inches in diameter for most convenient grasp. Conversely, they should not be so small as to require manipulation with thumb and forefinger.

Force limitations are important. A maximum torque of two inch-pounds is suggested for rotating knobs. Often an equipment will include frequently-used rotating controls with sharply milled surfaces, requiring such excessive torque as to result in chafed and abraded finger-surfaces in short order.

Toggle devices, such as on-off switches, should provide an easy visual cue to the switch position. Thirty degrees either side of center is usually satisfactory. Mount toggle switches so that the ON position is always forward, up, or to the right. Bar or pointer-shaped controls should be used exclusively for functions which have mechanical detents to aid positioning. Knobs which are less than three-quarters of an inch in depth should be knurled rather than serrated. For knobs greater in depth, serrations are adequate but should be chosen to give the best gripping characteristics.

Inadequate drip-proofing is a common defect of electronic equipment intended for Naval use. This is

* "Human Engineering Guide for Equipment Designers," Wesley E. Woodson, University of California Press, 1954.

an especially important factor because of the presence of spray, high humidity, and condensation in Naval vessels. Although an enclosure may divert the better part of incident spray, or protect the equipment against most of the drip from an overhead pipe, a sufficient quantity of moisture can be entrapped to cause eventual failure of the equipment. The eventual failure is almost certain in the case where the entrapped moisture is saline (for instance, seawater). Specific measures which will generally improve poor drip-proof performance are as follows:

1. Tubular gaskets, or exuded gaskets which have a tubular sealing surface, should be used. Flat gaskets are generally unsuccessful.
2. Bearing load of access doors should be carried by metal-to-metal contact, rather than through a gasket.
3. Where equipment is not required to be actually submersion-proof, adequate drainholes should be provided in both chassis and enclosure.
4. To prevent entry of water, louvres should be provided with baffle-plates, or a second set of inverted louvres should be used to back up the exterior louvres. In this way, moisture is forced to follow a lengthy and tortuous route if it is to enter the enclosure.
5. Proper seals should be provided around all outside hardware and components where moisture may enter.

Excessive line-conducted and/or radiated electrical noise is a common shortcoming of military electronic equipment. In Naval use this defect is particularly important. Such excessive noise, in the crowded confines of a Naval vessel, may compromise or disable other sensitive electronic devices. It is also possible that such radiations may be detected by enemy surface vessels, submarines, or aircraft.

Permissible levels for these effects are given in Military Specification MIL-I-16910A (SHIPS). A late change (Amendment No. 2, 1 November 1957) modifies the basic specification to require the determination of an "equipment spectrum signature", which is a presentation showing all radio-frequency radiations (both desired and undesired) of electronic equipments. Purpose of the "signature" is to provide in simple and graphic form a representation of the noise-generating characteristics of the equipment, which can be matched with characteristics of other sensitive equipment so that trouble can be predicted and guarded against.

In these times of high-performance electronic equipment, high power levels, complex spectra, and high orders of sensitivity are increasingly common. This means that the requirements for suppression of noise-effects almost approach those for water-tight integrity. First, and most important, is the necessity for a good line filter to prevent undesired radiations from reaching the power line. All circuits which leave

Official U. S. Navy Photograph

Figure 7. Walk-in Environmental Chamber & Control Panel.

the enclosure (keying lines, microphone lines, control circuits, etc.) must be similarly protected. RF weatherstrip should be used extensively around all mating edges where RF energy might escape, and large holes in the enclosure (meter holes, etc.) shielded wherever possible. Conducting glass in meter-faces often cures leakage from meter-holes. Extensive instrumentation is necessary for conducting actual measurements of noise fields, since the frequency range of interest extends from 15 kcs. to 10,000 mcs.

Poor surface preparation of metals is manifested by corrosion, blistering, and peeling of paint. Growth of fungus, and galvanic action between dissimilar metals in contact is promoted by improper surface preparation. *All metal surfaces should be properly degreased and cleaned before the prime coat is applied.* After cleaning, extreme care must be used in handling metal surfaces. The Laboratory often observes corrosion patches in the shape of a greasy thumbprint. Using care in cleaning surfaces, and adherence to the finish requirements of Specification MIL-E-16400B (SHIPS), will in practically all cases produce satisfactory results.

Poor wiring design is a frequent cause of trouble. This category includes insufficient slack in conductors, inadequate securing of cables, long unsupported runs of wiring, poor design of cable-harnesses, etc. In many instances these faults will not be obvious, but by the time an equipment passes through shock, vibration, temperature, and humidity extremes, what may have been a minor and unapparent shortcoming can develop into a major breakdown.

Spurious responses are a frequent cause of poor performance. This category includes parasitic oscillations, hum, regeneration, harmonic responses, and similar undesirable effects. As might be expected, spurious responses are usually aggravated by high line voltage.

Poor performance at extremes of temperature is another failing which is common. In contrast to shock and vibration effects, this category does not usually result in immediate and spectacular failure. Instead, it is manifested by slow frequency drifts, falling off of output power, poor audio fidelity, freezing of bearings, differential contraction of materials with resultant mechanical stress, and other marginal but inevitable consequences. The performance often becomes satisfactory again when the ambient temperature is returned to normal.

Examination of equipments with these symptoms usually shows that blowers, baffles, and louvres present for cooling purposes are (if they are present at all) added by the designer as an afterthought. But with the advent of miniaturization, and service requirements for even higher operating temperature, this "afterthought" technique is just not good enough. Cooling must now be considered as a prime design requirement; electronic equipment must be *thermally* designed as well as *electrically* designed.

The magnitude of the problem is suggested by the fact that many heat-dissipating parts in electronic equipment were found to operate *in excess of their rated temperatures* even when the equipment as a whole was subjected only to an ambient of 25°C. It is not hard to imagine what happens when the ambient temperature is raised to the Class 4 shipboard operating limit of plus 50°C. Vacuum tube envelopes are particularly susceptible to such elevated temperatures. Use of the new heat-dissipating tube-shields is often effective in reducing these temperatures to a safe level if allowable plate dissipations have not been exceeded.

There are many more categories of failure in shipboard electronic equipment under the evaluation process, but they will not be gone into in detail here. Inadequate insulation, loose fastenings, poor assembly, inadequate fusing—all these items take their toll, but in most cases the remedy will be apparent to the designer. And although the foregoing discussion has dealt generally with shipboard electronic equipment, it is obvious that many of the criticisms are equally applicable to other types and kinds of equipments intended for military applications. The shock and vibration problem, for instance, is common to all kinds of equipments and devices.

These foregoing categories of failure are significant because they are the summation of defects observed at the Navy Electronics Laboratory over a long period of time. In all cases, the reasons for failure have been communicated to the equipment designer, either in terms of a simple statement of failure where the reason and remedy for the failure were obvious, or by a more detailed analysis and a suggested remedy where the mechanism of the failure was obscure. In this regard alone, the evaluation program has much more than paid for itself. But there are collateral benefits beyond this which should be pointed out.

Briefly, the Navy has benefited from the exposure of the evaluation engineers and scientists themselves to the continuing best efforts of commercial designers. Evaluation engineers, engaged in day-to-day evaluation operations of significance, can not help but become wiser and more adept themselves at design and development as a consequence of seeing both the successes and the failures of the nation's military electronic designers. Consequently, in addition to the fact that they thus become better-qualified evaluators, they also are inevitably stimulated to suggest and carry out informed designs of their own. The Laboratory has a long and extensive background in the evaluation of radio-communication equipment, and in consequence the high-performance VLF receiver of Figure 9 was designed by engineers in the Evaluation Branch. Compact, rugged, reliable, and simple, it has been successfully subjected to much more severe treatment in shock and vibration than the Navy demands. Fleet installations of the VLF receiver have already elicited high praise. This, and other developments now under way, indicate the extent of an important collateral benefit arising from the evaluation process.

So far, we have been talking almost exclusively of *performance* evaluation of military electronic equipment. For many years this has been the chief objective of programs such as exist at the Navy Electronics Laboratory and elsewhere. In effect, what we have

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Figure 8. Low Temperature Testing.

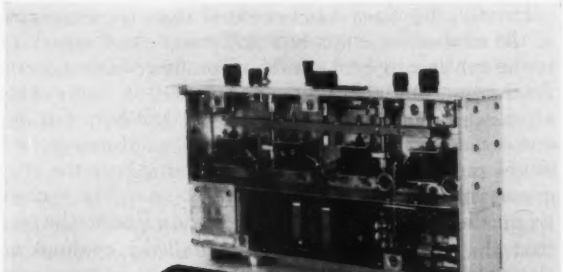

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Figure 9. VLF Radio Receiver.

been talking about are the electrical and mechanical parameters which can be determined on a short-term basis. Typical are bandwidth, sensitivity, noise figure, pulse width, shock and vibration characteristics, transmissibilities, temperature and altitude effects. But in recent years there has been increasing (and justified) attention turned to two other parameters, general in nature and difficult to deal with because of lack of quantitative standards. These are *reliability* and *maintainability*.

The first of these has to date been the subject of more attention than the second. Although reliability has always been a problem with electronic equipment, and has widely been recognized as such, it was in the same category as the weather; everyone talked about it, but no one did anything about it. There was good reason for this. It is difficult to take constructive action about reliability until there are some quantitative standards established for it. Until you can define it, in the first place, and measure it, in the second, little is apt to be done.

What is reliability? Almost anyone can give you his own definition. To the engineer it means one thing, to the manufacturer another, to the task force commander it has a third and different meaning. However, pioneering work by the Advisory Group for Reliability of Electronic Equipment (a committee established by the Secretary of Defense) and other agencies has brought about a definition which is now generally accepted. Reliability is the *probability* that an electronic equipment, having a statistically-determined Mean-Time-Between-Failures, will perform satisfactorily for a given period of time. Expressed mathematically, this relationship is

$$R = e^{-t/m} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where R is the reliability, e is the base of the mathematical system of natural logarithms, m is the Mean-Time-Between-Failures, and t is the mission time (or time during which the equipment is required to operate without failure).

In expressing R this way, it is obvious that the limits of R are from 1 to 0. A value of 1 for R indicates that operation for the period of time t is a certainty. Conversely, a value of 0 for R indicates that there is no possibility of the equipment operating satisfactorily for the period t. From an examination of the relationship, it can be seen that if t is small with respect to m, then R (the probability of successful operation for the period of time t) is large. If t is large with respect to m, then R is small. As a point of interest, when t equals m the probability R is equal to 1/e or 0.37, which means that there is about one chance in three of the equipment performing satisfactorily for the time t.

Of particular interest to us is the quantity m. This is the Mean-Time-Between-Failures. It is not to be expected that a dozen radio receivers, for example, will all display the same time-to-failure when they are put under test and run for a period of time. In general, one can expect a Poisson distribution of the times-to-failure. This is what makes the problem difficult. The only way to get any place with this problem is to determine *statistically* the Mean-Time-Between-Failure for this type of radio receiver. Unfortunately, this means also that you can have little confidence in your findings unless you have a reasonable-sized sample (desirably a half-dozen or so) and can run them long enough to get data which permits you to establish a mean with some accuracy. Too small a sample, or limited time of running, will give you an answer, but the answer may have such broad confidence limits that it is of little use.

Here is the immediate field where evaluation engineering can and will take on an important and tremendous burden. The MTBF will have to be determined for each new item of electronic equipment. The MTBF will become as important and significant as any other parameter, which is the way it should be. Only by detailed and comprehensive measurements can the MTBF be determined. Various methods have been proposed for *calculating* the MTBF during the process of development of an equipment, but these methods are at present very imperfect and unlikely to be substantially improved. In any case, the actual *measured* MTBF is desirable.

What are the advantages of knowing the MTBF? First, the using service can *know* the reliability of its electronic equipment in terms that are physically significant. A radio receiver with a MTBF of five thousand hours is a good receiver from the reliability standpoint. Secondly, the MTBF can be directly compared between different types of radio receivers, and techniques which made the MTBF of one very high can be profitably turned to use on other receivers, thus improving the performance of all of them. Third, the MTBF is a convenient and easily-understood number which can be used to improve the gen-

eral, across-the-board reliability of *all* equipment. For instance, if a sonar set has an MTBF of three hundred hours, is there any good reason why some radars consistently give only a hundred hours? Perhaps there *is* a good reason, but until the MTBF was invented, it was difficult to ask such a question; even to be aware that the problem existed in concrete and specific form. Maybe the radar designers need to take a lesson from the sonar designers. Maybe there are also techniques common to radar which are of value in designing sonar equipment. The MTBF can be a valuable link between such areas.

Maintainability is now coming in for the same share of attention as reliability. Here we have the same problem: what *is* maintainability, and how do you measure it? Many definitions of maintainability have been proposed, but there is still disagreement. A Navy Electronics Laboratory scientist has proposed the following definition: *

$$M = \frac{24 - (M_s + M_c)/91}{24} \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

where

M = maintainability

M_s = Total net down time for scheduled maintenance tasks over a 91 day interval (measured in hours)

M_c = Total net down time for corrective maintenance tasks over a 91 day interval (measured in hours)

For a typical Navy radio transmitter (AN/WRT-2) this type of calculation gives a value of M of .846, which indicates that the transmitter is available for service about 85 percent of the time.

Another index has been formulated to indicate the probability that any one malfunction will necessitate a given number of repair hours.** This approach yields a mean repair time of 9.4 hours for failures in the AN/WRT-2 equipment.

A third approach † derives an index of maintainability based entirely on repair time. This approach expresses the average repair time of an equipment in manhours, and states it as a reciprocal to make the maintainability increase as the mean net time-to-repair-failure decreases.

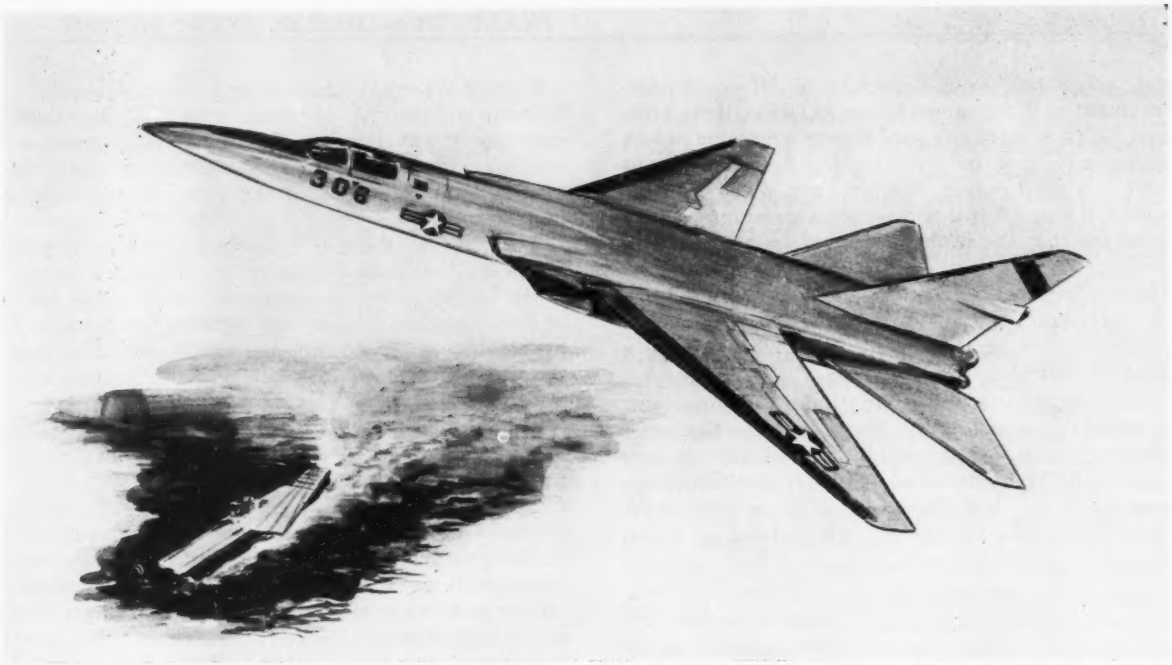
These various approaches are cited to show the different kinds of thinking on the subject of maintainability. It is likely that the end result will be some kind of maintainability index which takes the form of a probability; that is, a number which indicates that the mean time to repair a failure will lie within certain limits.

The relationship between reliability and maintainability should be kept in mind. If the reliability (as previously expressed) is 1.0, then there is no maintainability problem. Conversely, if the reliability is 0.0, then the whole problem is one of maintainability. In the practical case, the reliability lies somewhere between these limiting values, and the maintainability problem is more or less important depending on the reliability.

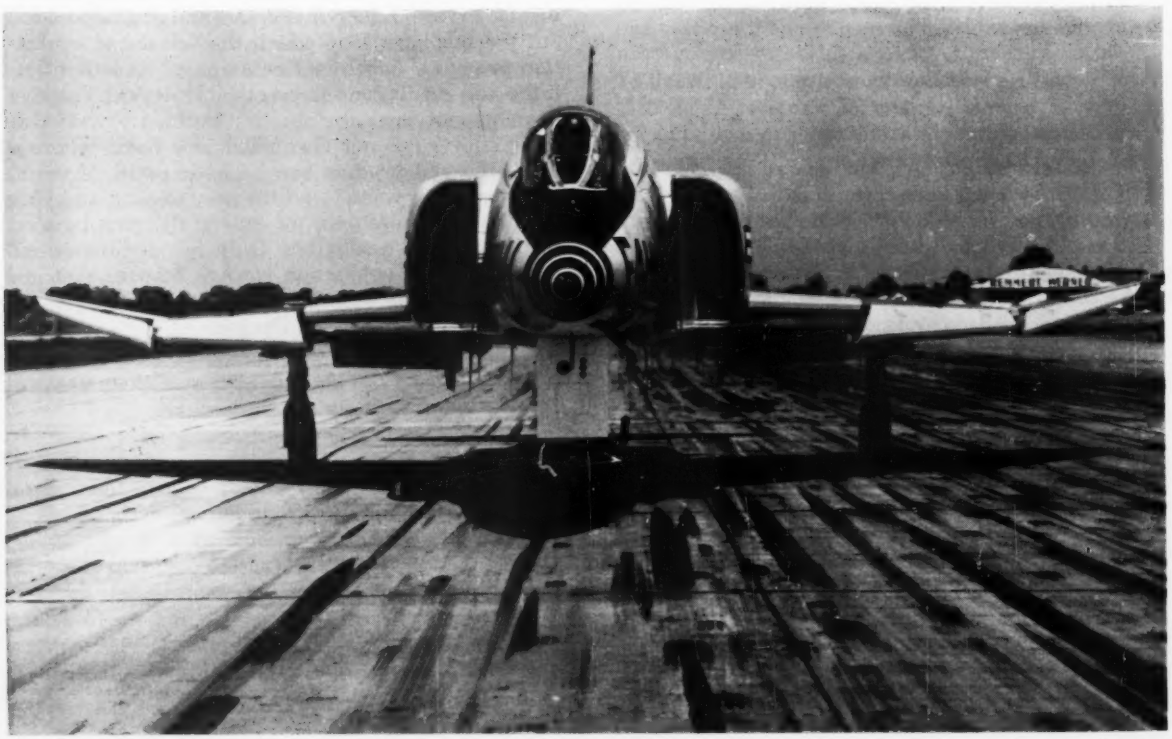
In retrospect, now, perhaps all this seems a long way from the fable of Oog and Guk which began this discussion. But the principles involved are the same. Research and development are a necessary and indispensable part of our modern life, and particularly are they necessary to the continuing defense of the free world against aggression. But research and development are largely ineffective and unguided without the feedback loop which the science of evaluation provides. A brilliant research effort is fruitless if the end result is an impracticable device. The best development engineers in the world are wasted if their efforts are not channeled into fields where a workable end product can be anticipated. If we as a nation don't want to go the way of Oog and Guk and the tyrannosaurus, we constantly must be alert in this field of evaluation. Only by continuing and comprehensive evaluation can we be sure that our weapons are the best. In the modern world there is no place for second-best weapons.

* E. S. Bean, NEL Technical Memorandum No. 242, 23 April 1957.
** ARINC Report "Maintainability of the AN/WRT-2 Transmitter," 14 Nov. 1957.

† AGREE Report "Reliability of Military Electronic Equipment," 4 June 1957.



North American's A3J is shown in this artist's concept. This aircraft is designed as the Navy's future carrier based, all-purpose, attack aircraft. This supersonic aircraft, the VIGILANTE, was announced in late spring. It made its first flight in August and is now undergoing a series of test flights.



The Navy's most recently announced fighter, the F4H, is shown poised on the runway in this unusual view. This McDonnell built aircraft is powered by two J-79 engines which are expected to give a speed of over Mach 2. An all-weather ship, it has a wingspan of 38 feet 5 inches and a length of 56 feet.

"THE SHIPBUILDER & MARINE-ENGINE BUILDER"

FREE-PISTON PROGRESS

ACKNOWLEDGEMENT

This article, published in the April 1958 edition of "The Shipbuilder and Marine Engine-Builder," is based on a paper delivered before The Institution of Engineers and Shipbuilders in Scotland. The authors were F. A. I. Muntz, of Alan Muntz & Co., Ltd., and M. Barthalon, of Société Industrielle Générale de Mécanique Appliquée (S.I.G.M.A.).

IT is proposed, first of all, to consider briefly the reasons why potential users have, during the past two years, shown such increasing interest in the free-piston engine system. It is probably due to two factors. First, a general feeling that the crankshaft engine has now reached a point where its further development has begun to involve operating conditions and complexities, which are rendering it too costly and troublesome. Second, the realization that the free-piston engine system combines the virtues of piston and gas-turbine engines and, moreover, in a way which eliminates their respective weaknesses, in particular the high bearing loads and crankshaft torsional vibrations of the conventional crankshaft engine and the special materials made necessary by the high temperatures at which gas turbines have to run to obtain an acceptable fuel consumption. These factors have, however, been reinforced by two comparatively recent findings, *viz.*, on the one hand that free-piston engines in service are, in fact, proving reliable and comparatively easy for unskilled personnel to operate, and, on the other, that the larger free-piston gasifier units can burn residual fuels, including bunker-"C," "as it comes" and without ill effects. As a result, the combination of free-piston gasifier and turbine is now beginning to be seen as a basically simpler and cheaper engine system.

Two sizes of free-piston gasifier have so far been developed for marine and industrial purposes—the GS 34, 1,250 gas H.P. unit in France, and the CS-75 420 gas H.P. unit in England. The GS-34 unit is now in production not only in France, but also in the United Kingdom, and some 100 units manufactured by S.I.G.M.A. have to date completed more than 250,000 hr. in actual service. The CS-75 unit is (at the date of the paper) about to go into selected service in the form of auxiliary 200-kW. sets. It incorporates some of the proved good features of the GS-34, but, basically, it conforms with British engineering predilections and, therefore, represents a complementary contribution to the free-piston art, not only in point of size, but also of engineering technique. Of course, the free-piston engine, although its moving parts appear comparatively simple and robust, possesses its own unique characteristics and troubles. These needed much test-development, first of all to discover and then to solve them. The following are typical examples: —

Internal-combustion Pistons.

These have to stand high maximum pressures of the order of 1,800 psi and, in addition, have to keep cool under high-duty conditions. Initially, attempts were made to develop pistons with hot crowns to improve thermal efficiency, but trouble was experienced both with piston cracking and with top piston rings breaking and wearing. However, the composite steel piston, developed for the GS-34, now provides good operating conditions for its piston rings and there is no trouble with piston breakage. The cast-iron piston developed for the CS-75 is of comparatively simple design, having radial ribs under the crown, with a "heat dam" consisting of a reduced wall thickness above the top ring and holes through the ribs at this point to avoid conduction of heat down the rib directly from the crown to the top ring. An interesting point in connection with this comparatively simple piston is that at full load of 420 gas H.P., the temperature just above the top ring is not more than 180° to 190° C. (356°-374°F).

Internal-combustion Cylinder.

The center section of the GS-34 combustion cylinder initially suffered from cracking around the injector bosses, usually between 1,000 and 3,000 hr. of running. This has been overcome by modifying the design to give better cooling and to prevent local thermal stressing, and by water treatment to prevent corrosion fatigue. The original US-75 prototype of the CS-75 experienced similar troubles with a conventional one-piece cast-iron cylinder. The present construction of the CS-75 cylinder, with a reinforcing ring shrunk over the central part of a one-piece cast-iron liner, has been completely trouble-free over some 10,000 hr. of running of the CS-75 prototypes, and this in spite of peak pressures of the order of 4,000 psi experienced during early experimental work with water-cooling grooves in the liner.

Engine-case Fires.

Initially, both the GS-34 and CS-75 prototypes experienced engine-case fires, which, while free from explosion risks, neverthless caused engine stoppages. These tended to occur with dirty engine cases when scavenge-air temperatures exceeded 220° to 240°C (428°-464°F). This trouble has been eliminated by careful elimination of hot surfaces (above 240°C) and oil pockets, the use of lubricating oils which are free from the tendency to form deposits in the engine case, and by watercooling the compressor-head plates and so reducing the temperature of the air delivered to the engine case.

Scuffing.

Both the GS-34 and CS-75 have suffered from three types of scuffing—that caused by swarf when running-in a new engine, that caused by non-rectilinear motion of the moving parts, and that by mal-distribution of the oil on the working surfaces. It has so far proved impossible to eliminate swarf from new engine cases and compressor pistons; but a routine of very short initial runs has made it possible to eliminate swarf before it causes damage.

Reactions of the synchronizing gear, slight misalignment, and slightly wavy cylinder surfaces, have all been suspected as a cause of occasional scuffing. But, given correct manufacture and assembly, better oil distribution seems to be the all-important factor. The latter has been obtained (*a*) by increasing the number of oil-feed points in the internal-combustion and compressor cylinders, and (*b*) by increasing the frequency of oil pulses, in the case of the GS-34 from about one pulse per 200 strokes to one pulse per 20 strokes, and in the CS-75 from one pulse per 200 strokes to one pulse per six strokes.

Lubrication.

Initially, it was not appreciated that lubricating oils would need to be developed to meet the following special conditions obtaining in free-piston gasifiers:—

(*i.*) Lubrication of the internal-combustion pistons and cylinders by metered feeds (total loss) to various points on the cylinder wall. The requirements are those for high-duty, very highly supercharged, two-stroke cycle Diesel engines with a high maximum cylinder pressure; and, in addition, the oil must not deteriorate or leave deposits in the small-bore pipes which pass through the engine case containing the scavenge air at temperatures up to 240°C (464°F).

(*ii*) Lubrication of the compressor cylinders, partly by metered feeds and partly by any leakage of oil from the synchronizing mechanism and from the piston cooling-oil circuit. The requirements here are easy from the point of view of lubrication, as the compressor operates at a quite low pressure, but the oil must not decompose in contact with a large excess of air and produce deposits on the surfaces of the delivery valves, or of the engine case at temperatures up to 240°C.

(*iii.*) Cooling of the internal-combustion pistons. A low viscosity oil is desirable to get good heat transfer to keep the underside of the piston crown and the top piston ring cool.

SMALL FREE-PISTON ENGINES

In addition to the two engines already referred to, development of several other free-piston engines has been proceeding during the past few years. Although the paper is not primarily concerned with them, the small free-piston engines should be mentioned as evidence of interest in this modern form of the free-piston principle for an increasingly wide range of applications.

The free-piston air-compressor really forms a different family from the gas-generator or "gasifier." These are now becoming fairly well known and are in use in portable form in fair numbers by contractors. There was first the French-made P-13 compressor delivering 200 cu. ft. per min. at 85 psi, of which there are about 1,500 in use, and this was followed by the smaller British-made P-42 compressor delivering 105 cu. ft. per min. at 90 psi. A recent development, the DAC 130, using direct-air flow ("streamline") valves instead of plate valves, but otherwise of the same dimensions, has allowed its rating to be increased to 125 cu. ft. per min. at 90 psi with a reduced specific fuel consumption.

Returning to free-piston gasifiers, there are four engines smaller than the CS-75 which are known to be under active development in the prototype stage.

LARGE FREE-PISTON SETS

Two large sets using GS-34's are now under construction in this country; a 13,000 H.P. gas compressing plant is being constructed by the Kellogg International Corporation for the Imperial Chemical Industries, Ltd., for use at Billingham (Figure 1), and a 36,000 kw. power station is being supplied by Associated British Engineering, Ltd., for Singapore, which consists of six 6,000-kw turbo-alternator sets and incorporating eight GS-34's (Figure 2). Two 16,000-S.H.P. propulsion sets are under construction for frigates for the French Marine Nationale. Each set has two propeller shafts, each driven by two double turbines supplied by eight GS-34 gasifiers. So far as marine applications are concerned, orders have so far been received by British licensees for five free-piston propelling sets. The first one was a 2,500-S.H.P. set for a new ore-carrier. Then there are two other new ships, a fast cargo liner of 7,000 S.H.P. and a deep-sea tramp of 4,000 S.H.P. and conversion sets for two cargo ships of 4,000 S.H.P. and 2,000 S.H.P., respectively.

When larger powers are required than can conveniently be supplied by simply increasing the number of 1,000-H.P. gasifiers, such as the GS-34, the limit of which may be considered to be in the region of 12,000-15,000 H.P., there are several possible ways of retaining the basic advantages of free-piston engines.

The first would obviously be to use larger gasifiers

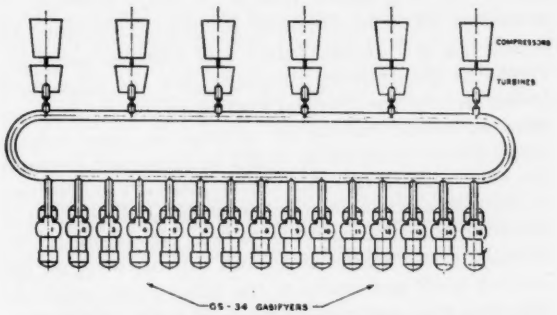


Figure 1—13,000-H.P. Compressor Plant, including Two Stand-by Units.

of, say, 2,000-2,500 S.H.P. This is a perfectly practicable size, and, in fact, designs for such an engine similar to the GS-34 have been worked out. However, a new engine like this would inevitably require the usual process of development, which would take some years before achieving full reliability and would cost a proportionally large sum of money. This development will no doubt be undertaken in due course, but, for the present, it seems better to concentrate on the application of the already developed 1,000-H.P. unit.

Another way to obtain more power from one unit would be to supercharge the standard GS-34 gasifier with a turbo-blower, preferably with an intercooler

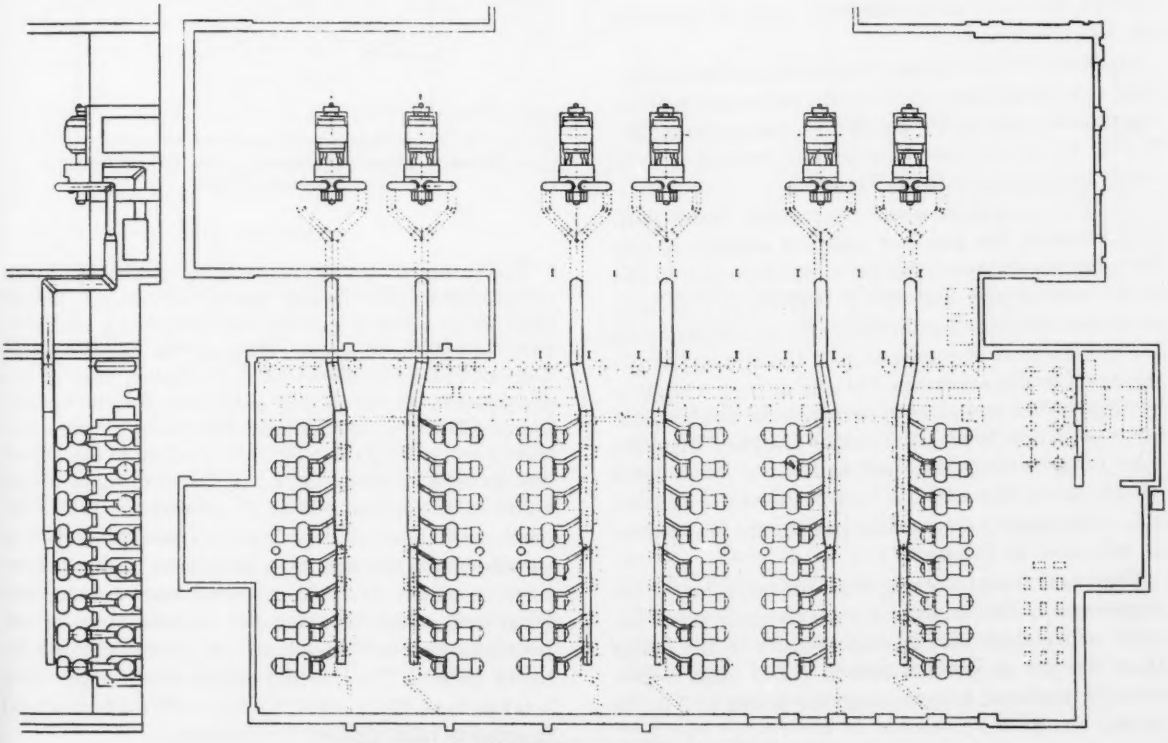


Figure 2—36,000-kW. Power Station for Singapore.

before the compressor intake. A large increase in power would be possible with a supercharge pressure to say 1½ to 2 atmos. (abs.) and a proportional increase in gas-delivery pressure; but the maximum pressures and the thermal loading of the combustion cylinder would inevitably increase to some extent, although not so much as might be supposed, because the compression ratio in the combustion cylinder of a free-piston engine can be controlled independently. An attractive alternative is to twin GS-34 units with an engine case common to both. In this way, all the moving parts and many of the other parts would be identical to those in the already developed GS-34.

MIXED GAS/STEAM PLANT

An altogether different way of obtaining higher power is the use of a plant working on the mixed gas/steam cycle. This would consist of a group of gasifiers supplying an expansion turbine, and a steam boiler supplying a steam turbine and making use of the gas (which consists of 800 percent unburned air) as highly preheated combustion air in which more fuel is burned, the feed water picking up the waste heat from the engine water jackets, etc. In this way, a power output about four times that given by the gas-generators themselves can be obtained at a very high overall thermal efficiency, higher, in fact, than either the gas cycle or the steam cycle separately, because of the ability of the steam cycle to pick up some waste heat from the gas cycle.

There are two rather different ways of carrying out this scheme, *viz.*:—

(*a*) With a normal-pressure-combustion boiler supplied with the exhaust gas from the expansion turbine at practically atmospheric pressure, and at about 250° C (482°F) at full load. The general scheme is indicated diagrammatically in Figure 3.

(*b*) With a high-pressure-combustion boiler situated between the gasifiers and the expansion turbine. In this instance, the gas supplied to the boiler is at about 45 psig and 450°C (842°F) at full load. After burning more fuel in the boiler and transferring most of the heat of combustion to the steam, the gas passes on to the expansion turbine at only a slightly lower pressure and at a temperature either about the same or 50° to 100°C (90°-180°F) higher than the inlet temperature. The heat in the gas leaving the expansion turbine and the heat in the water jackets, etc., of the gasifier are used for heating the feed water, as indicated in Figure 4.

The chief feature of the high-pressure boiler arrangement is that it gives a very compact plant because of the high rate of heat transfer in the boiler from the gas at about 4 atmos. (abs.) with a permissible pressure drop through the boiler of 2 psi or more.

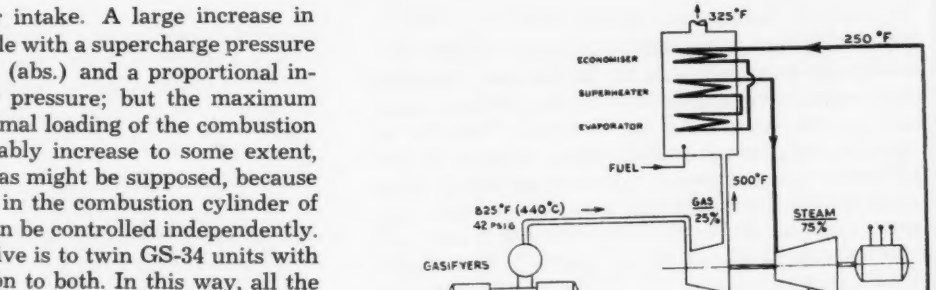


Figure 3—Mixed Gas/Steam Cycle, Normal-pressure-combustion Boiler.

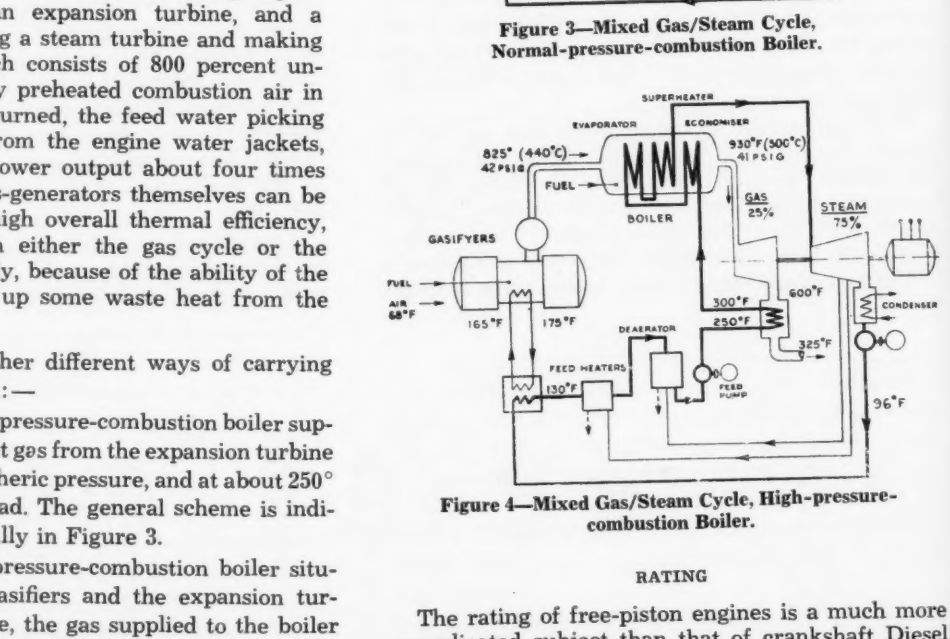


Figure 4—Mixed Gas/Steam Cycle, High-pressure-combustion Boiler.

RATING

The rating of free-piston engines is a much more complicated subject than that of crankshaft Diesel engines and has a number of interesting features. First of all, there is the rating of the gasifier itself, without regard to the turbine it supplies, that is, the gas-delivery pressure and mass flow can be chosen independently of each other. The rating is then primarily a matter of the maximum loading of the Diesel pistons for good endurance, and the limiting practical value of the piston stroke in relation to the outer mechanical limit of their travel. There may also, in certain circumstances, be a limitation to the gas-delivery pressure to avoid excessive engine-case temperatures, which would cause carbonization of the lubricating oil on its walls and in the compressor delivery valves. The limiting engine-case temperature is taken to be 240°C (464°F). This will evidently tend to occur at high ambient temperatures.

Although the gas-delivery pressure is not in itself a limiting factor and could be chosen over a substantial range of from, say, 40 to 50 psig, a number of factors tend to narrow the choice considerably towards the middle of this range, and, for the purposes of a standard rating procedure for the GS-34, it has been taken as 57.5 psia or 42.8 psig at standard atmospheric pressure.

With regard to the maximum piston loading, this is practically determined by the output of the gasifier in gas horsepower. In the case of the GS-34, the continuous maximum rating is now established at 1,250 gas H.P., corresponding to a nominal 1,000 S.H.P., as a result of extensive running experience. Outputs of well over 1,500 gas H.P. have been sustained for considerable periods of time on test, so that the continuous rating may be considered conservative. The continuous maximum rating of the CS-75 has provisionally been set at 420 gas H.P., corresponding to, say, 350 S.H.P., but there has not yet been sufficient running experience to confirm this value.

The maximum stroke must obviously be given an O.D.P. far enough back from the outer mechanical limit to allow for the overstroke trip to be set at a reasonable distance from it, and to allow for the stroke-to-stroke variations in outer deadpoint, so that the overstroke trip does not normally operate. In the case of the GS-34, with a mechanical limit of 550 mm. (21.6 in.), the normal maximum O.D.P. is established at 485 mm. (19.1 in.), that is, 88 percent of the stroke between mechanical limits. With a normal full load I.D.P. of about 30 mm., the stroke is 2 x 455 mm. (2 x 18.0 in.).

DE-RATING.

Turning now to complete free-piston gasifier sets including the turbine, and the question of de-rating for non-standard ambient conditions, there are two distinct cases to be considered:—

(*a*) *Adjusted Turbine.*

The nozzle area of the turbine is chosen or adjusted to suit the particular ambient conditions being considered. This normally applies to the case of choosing the normal ambient conditions at which a set with a turbine, of the usual axial type with a fixed nozzle area, should be rated for any particular application; or, of course, it would also apply to the derating of a given set to a turbine with adjustable nozzle area, such as a radial inflow turbine with variable nozzle blades, which might be developed for the purpose.

Fixed Turbine.

The nozzle area is fixed to suit one particular set of ambient conditions that differ from those being considered. This is the normal case to be studied when working out the derating of a given set with the normal type of axial-flow turbine with a fixed nozzle area.

These two cases are different because, with the adjusted turbine, the gas-delivery pressure can be maintained at its optimum or maximum value, whereas, with the fixed turbine, the gas pressure is determined by the delivery of the gasifier and the fixed swallowing capacity of the turbine. It is clear that the extent of de-rating will be smaller with an adjusted turbine than with a fixed turbine, both for a rise in ambient temperature and for a fall in ambient pressure, due to either altitude or barometer: in fact, it is a great deal smaller, only about one-fifth for ambient temperature and about one-third for ambient pressure.

It should be emphasized that the output of the GS--34 gasifier is not limited by the combustion in the Diesel cylinder, that is, by the trapped air to fuel ratio, which is always more than about twice the theoretical; so the basis for the derating is entirely different from that of a normally aspirated Diesel engine.

Although a fixed-turbine set is not better as regards de-rating than a normal Diesel engine, and in some respects rather worse, it is the two factors mentioned herewith which confer the great advantage on the free-piston gasifier set of a very small amount of de-rating when the nozzle area is chosen or adjusted to suit the normal operating conditions of the set.

OPERATING EXPERIENCE WITH FREE-PISTON GASIFIERS

The two small cargo vessels *Cantenac* and *Merignac*, equipped with two free-piston GS-34 gasifiers, have been at sea for four years. Table I summarizes their performance up to the 1st October, 1957.

TABLE I.

	Cantenac.	*Merignac.*
Years of service	4	3½
Miles travelled	145,000	120,000
Hours at sea	18,000	15,500

Both ships achieved the best record of regularity of their fleet. Once a year, each is berthed for a period of two weeks, in order to carry out general ship's maintenance. Throughout the rest of the year, the engine-room personnel does all minor maintenance work without ever exceeding the ship's scheduled periods in port.

On the 1st October, 1957, there were 104 GS-34 gasifiers in service in nine different countries in different parts of the world. Apart from the 42 delivered to the French Navy, most of the gasifiers average between 500 and 700 hr. of operation per month; therefore, the amount of operating experience is increasing very quickly. Each of the first 15 have now achieved between 10,000 and 17,000 hr. of operation; this has provided the builder with a sound basis to establish wear rates and long-term maintenance rules. With due respect to the remarkable ability of users to invent troubles unknown to engine-builders, it can be said that the variety of operating conditions and supervisory personnel is such that most of the possible

troubles have now been discovered, and that free-piston machinery based on the GS-34 gasifier is today reasonably foolproof.

"William Patterson"

The conversion of the *William Patterson* is part of a program devised by the U.S. Maritime Administration to obtain operating experience with four converted "Liberty" ships using different types of powerplants: steam turbine, Diesel engine, open-cycle gas turbine and free-piston engine. The contract for the free-piston machinery for the *William Patterson* was awarded to the General Motors Corporation in October, 1954, and its fulfilment was the result of very close team work between General Motors and S.I.G. M.A.-S.E.M.E. of France. The installation and conversion work was done by the Bethlehem Steel Company at their Baltimore Shipyard. As shown in Figure 5, there are six GM-14 gasifiers (practically identical to the GS-34) delivering gas into a common manifold which feeds two reversing turbines, each having two gas inlets.

The decision to use two turbines with twin inlets was arrived at for three reasons:— First, it was thought prudent to make a limited jump in size from the 1,000 S.H.P. reversing turbines previously used; second, the scheme gave the maximum flexibility in operation and testing, which was especially desirable in this first large free-piston ship, which has to provide basic data for future vessels; third, the arrangement of the reduction gear was somewhat similar to a steam-turbine gear, with input shafts from h.-p. and l.-p. turbines.

The use of reverse wheels in the turbines, instead of a reversing propeller, was adopted as a result of the very satisfactory operation of the first 23 free-piston ships using 46 turbines with internal direct-reversing wheels. This had been proved by experience to be an extremely simple and entirely reliable scheme. However, it does not give as much reverse power as a reversible propeller.

Power control and reversing control are achieved hydraulically from a console, through the use of a single handwheel. The same console provides all the indications necessary for the remote control of gasifiers, turbines and auxiliaries. Any trouble can therefore be immediately located. The controls can, alternatively, be directly operated from the bridge.

FREE-PISTON ENGINED ORE-CARRIER.

A new ore-carrier now under construction by Messrs. Lithgows, Ltd., consists of three GS-34 gasifiers, on order with British Polar Engines, Ltd., for the Free Piston Engine Co., Ltd., supplying a turbine manufactured by Messrs. Rankin & Blackmore, Ltd. This is the first British-built ship to have free-piston machinery.

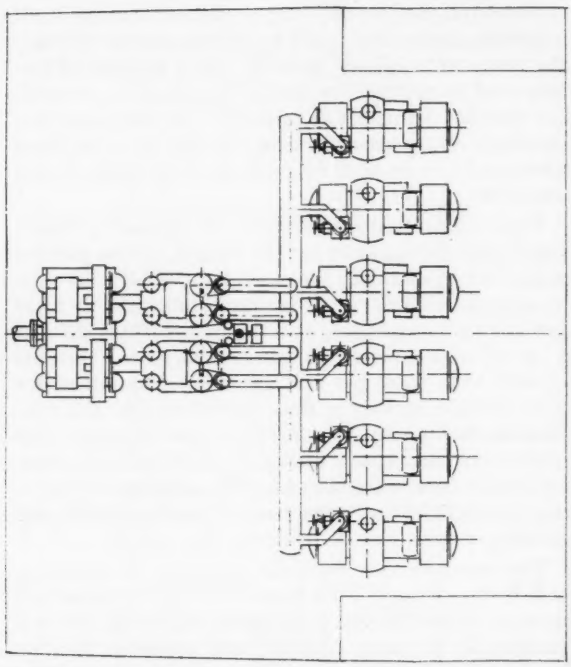


Figure 5—Arrangement of Free-piston Propulsion Machinery in the "William Patterson"; Plan View at Lower Level.

MARINE AUXILIARY SETS.

The CS-75 gasifier appears to be very suitable for marine auxiliary electrical generating sets, either as a single-cylinder engine producing 200 kw., or with two CS-75's supplying one turbine and producing 400 kw.

The 200-kw. sets are under construction. Various arrangements of the plant are possible, but the overall dimensions of the units on order are 10ft. 6in. by 9ft. 6in. by 7ft. 9in. high, and the weight is 5 tons. Preliminary tests indicate that the CS-75 will run satisfactorily on residual fuel of 960 sec. Redwood No. 1 at 100°F, and it is hoped that it may be capable of burning the same fuel as the main engines.

CONCLUSION.

It is hoped that sufficient has been included in the foregoing to show that the free-piston gasifier/turbine-engine system has become a practical proposition and, moreover, one deserving serious consideration for a wide range of applications. With this in mind, the authors have drawn attention to the features which have given trouble, in order to emphasize not only that satisfactory solutions have been found, but also that experience has now shown that the free-piston engine system does not, in fact, suffer from any basic insoluble troubles of its own. Admittedly the free-piston engine has taken a long time to

emerge, but apart from the delay caused by the last war, there are two major reasons for this. Firstly, the free-piston gasifier/turbine-engine system is the logical final stage in the development of the internal-combustion engine, and so it has had to wait for "turbo-charging" to show up crankshaft limitations and provide the necessary stepping-stones. Secondly, it is often impossible with a free-piston engine to alter only one thing at a time, that is, eliminate one "bug" at a time, and so it has had to go through a long period of trial and error to establish the exceptionally extensive but all-important know-how.

Russian representatives to the Geneva Conference disclosed new information concerning the nuclear propelled icebreaker, LENIN. The total shaft horsepower of the ship is 44,000. Three pressurized water reactors will provide steam for the turbo-electric propulsion plant. Each reactor is furnished with an independent primary circuit and steam generator. The three reactors will supply steam to four turbo-generators of 11,000 HP each, which in turn will provide electric power to the three propulsion motors driving the three shafts. The LENIN will have a cruising speed of 18 knots in open water, and it is estimated that her endurance will be one year's steaming.

Japan presented a design proposal for a nuclear powered submarine tanker at the Geneva conference. The submarine, having a length of 590 feet and a displacement of 48,200 tons, is designed to carry 30,000 tons of deadweight cargo at 22 knots submerged. The proposed pressurized water reactor would supply steam at about 480°F and 565 psi to two propulsion units. Arranged with most of the cargo oil outside the pressure hull, the safe operating depth is designed to be 328 feet.

—from NUCLEONICS
September, 1958

"THE MARINE ENGINEER & NAVAL ARCHITECT"

WATER-LUBRICATED BEARINGS FOR MARINE USE

ACKNOWLEDGEMENT

This is an abstract of a paper read before the North-East Coast Institution of Engineers and Shipbuilders at Newcastle-upon-Tyne on 31 March 1958, as published in the May 1958 edition of "The Marine Engineer and Naval Architect." The author is A. D. Newman, of Pametrada.

AT FIRST sight, water does not appear to be a very suitable lubricant; in fact, it does not seem to be a lubricant at all: it is not "oily." On the other hand, the effect of a film of water on a smooth road in reducing tire friction is well enough known, so that it must be realized that the concept of "oiliness" as a criterion of lubricant suitability, unless carefully defined, involves a misleading mental picture. A difficulty is that no general agreed definition of "oiliness" exists, but a useful though over-simplified concept may be obtained by considering it to be the property of a fluid which produces a reduction of friction between two rubbing surfaces. It is not an inherent property of the fluid *per se*, but one of the combination of fluid and bearing material, and is often considered to be largely or entirely synonymous with that other property of fluid/surface combinations called "wettability."

Since water possesses viscosity and density, it may support hydrodynamic lubrication in the same way as oil. Unless, however, it and the surfaces involved also possess "oiliness" in the considered sense of the word, then very little practical use can be made of its hydrodynamic propensities, because all practical bearings must operate on occasion under non-hydrodynamic conditions, for instance when starting and stopping, where this property is the one that matters. It is well-known that there are some materials which are not satisfactorily lubricated by oil, in general those having surfaces which are not "wetted" by oil. For oil lubrication, therefore, although of course other factors are involved, materials such as whitemetal, which give a good lubricant/surface condition, are in common use. Practical water lubrication thus depends on materials which will give similar results with water, and the property required may be investigated by tests under non-hydrodynamic conditions, that is low-speed tests at high loads. For this reason a considerable amount of work has been carried out at Pametrada on the low-speed performance of various materials with water lubrication.

LOW-SPEED WATER LUBRICATION

The test machine used in this work consists of a shaft driven by an electric motor through change gears, giving speeds between 30 and 3,000 r.p.m., on which is carried a pad of the material whose performance is being examined, the pad being of 60 degrees arc and loaded downwards onto the journal by dead weights and a lever arm. Water is supplied at a pressure of 2 psi to the leading edge of the bearing pad. Typical tests, and results obtained with this machine, were as follows:

Conditions: Journal surface speed 0.9 ft. per sec. Fresh water.

Test: Measure static coefficient of friction over a range of loads.

Run for two hours at a specific load of 100 psi, then measure static and running coefficients of friction and examine bearing and journal.

Run for six hours at 100 psi load and examine bearing and journal.

Work up load to 400 psi in steps of 50 psi, running for one hour at each increment of load.

Run for 50 hours at 400 psi load and then examine bearing and journal.

Results: Material (a)—fabric-reinforced phenolic resin—operated without trouble over the range of loading. Its static coefficient of friction remained approximately constant at 0.35 (range of load 0-500 psi) during the test. The running coefficient of friction was approximately constant over the range of load involved but fell slightly during the course of the test, from 0.25 to 0.22. Up to a load of 400 psi no significant wear of journal or bearing was observed; during the 50-hour run at 400 psi the bearing wore 0.008, 0.006, and 0.002 in. at its leading edge, center, and trailing edge respectively. No measurable journal wear was observed.

Material (b)—asbestos-reinforced phenolic resin—operated without trouble over the range of loading. Its static coefficient of friction varied between 0.29 and 0.36, and its running coefficient of friction fell from 0.215 to 0.18 during the course of the test. No significant wear of journal or bearing occurred up to a load of 400 psi, but during the 50-hour run at this load, although no measurable wear of the journal occurred, the bearing wore 0.009, 0.0085, and 0.005 in. at its leading edge, center, and trailing edge respectively.

The way in which these bearings wore indicated non-hydrodynamic operation, and under this condition the rates of wear were relatively low. The results therefore show that if the right choice of bearing material is made, good non-hydrodynamic operation may be obtained with water, allowing practical water-lubrication to be considered seriously. The hydrodynamic performance of water then becomes of major interest.

HYDRODYNAMIC WATER LUBRICATION

The relevant properties of water are shown in comparison with those of a typical marine-geared-turbine lubricating oil in Table I. From this table it may be seen that the viscosity of water at normal temperature is much lower than that of oil. Its specific heat, however, is more than twice as great and its density is a little higher.

TABLE I
Properties of Water and Lubricating Oil

	Water	Typical turbine lubricating oil
Viscosity (centistokes)		
at 100 deg. F.	0.8	90
at 200 deg. F.	0.28	11
Specific gravity	1.0	0.89
Specific heat	1.0	0.45

The effect of a low-viscosity lubricant that is most important is that on the minimum lubricant film thickness in the bearing. For a journal bearing, for example, other things being equal, the minimum film thickness in practice appears over a small range of variation to be approximately directly proportional to viscosity and speed; in other words, for the same life of the journal with two different lubricants the ratio of speeds must be the inverse of the ratio of viscosities. With water lubrication this emphasizes the importance of non-hydrodynamic operation, because it means that under the starting and stopping condition non-hydrodynamic operation will persist over a much greater speed range than when oil is the lubricant. It should be observed at the same time that the ratio of viscosity between oil and water reduces as temperature increases, so that the disadvantage of water in this respect is less as the temperature level in the bearing increases, i.e. as speed increases.

There is, however, a further factor which becomes more and more important as speed increases, and this is the bearing heat balance; the design of high-speed bearings is dominated by the heat generation and need to provide adequate cooling. With most bearings there occurs a vicious circle effect, in that the removal of heat is effected by the medium whose shear is generating the heat; the greater the flow of lubricant through the bearing the greater may be the hydrodynamic loss in the bearing so that there is in many cases a practical limit to the speed at which the bearing may be operated satisfactorily. This limit depends on the relation between the capacity of the lubricant as a coolant and that to give hydrodynamic loss, the former being a function of specific heat and density and inversely of viscosity (since this has a considerable effect on flow) and the latter mainly of viscosity. It may be seen, therefore, that the limit with water is higher than that with oil.

Thus there is no inherent reason against the use of water as a hydrodynamic lubricant, though compared with oil for equivalent conditions a higher speed is required for the same load-carrying capacity. The considerably greater effective cooling capacity of water is a major advantage in favor of water.

PRACTICAL WATER LUBRICATION

The two essential elements of practical water lubrication are the ability to accept non-hydrodynamic or rubbing conditions without trouble, and the ability to provide hydrodynamic lubrication, or to "float" the journal or thrust collar, at speed, without contact between the solid parts of the bearing. The first element depends mainly on the properties of the combination of lubricant and surface, and requires particular bearing materials for suitability with water in the same way that particular though different bearing materials are required for suitability with oil, or any other lubricant. The second element depends mainly on the inherent properties

of the lubricant and on the geometry and conditions involved in the bearing.

STERN-TUBE AND A-BRACKET BEARINGS

It is common practice to use water as the lubricant in stern-tube and A-bracket bearings. Their design is largely based, however, on the bearing material used rather than on hydrodynamic grounds. By far the most common bearing material in this application has been *lignum vitae*, which possesses excellent properties under the conditions of water lubrication, but, due to the dimensions of the tree which is its source, is limited in size and dimensions. These bearings, therefore, are made as long staves of *lignum vitae*, dovetailed or locked into a housing, with very small water grooves between them. Since the uninterrupted circumferential length, or arc, of each stave is relatively small the possibility of appreciable hydrodynamic load-carrying capacity is remote, and the bearing must be made very large, particularly as regards its axial length, in order to keep the specific load to a very low value, a typical figure for this type of bearing being 7 psi.

The great axial length of stern-tube and A-bracket bearings of this type has two major disadvantages. A very small misalignment between bearing and shaft, which cannot be avoided in a relatively flexible structure such as a ship, especially when variation in loading may occur, has a considerable effect, and it is not uncommon experience to find a stern-tube bearing, for instance, with signs of loading over perhaps only a third or less of the total length. Secondly, the longer the bearing the less is the likelihood of the whole length being fully lubricated and cooled by the water, and again it is not uncommon to find the forward end of a stern-tube bearing charred, or sludged up, or otherwise ineffective.

Other materials than *lignum vitae* are now being introduced. Rubber and reinforced phenolic resins are perhaps the best known, but the possible range is wide. These materials in general are consistent and are not restricted dimensionally as is *lignum vitae*, but they are being used in most cases to replace it directly, in designs limited by the availability of natural wood. The result is that these new materials do not show to their best advantage; the bearings still suffer from the basic faults of excessive length and small arcs of surface, which are not corrected by a simple change in material, and in some cases trouble occurs due to these faults.

The greatest advantage of these materials lies in the fact that they are not nearly so restricted dimensionally as *lignum vitae*. Specific loads very much higher than those used in practice may be accepted by these materials under non-hydrodynamic conditions, i.e. at low speeds and starting and stopping, and this enables the bearing length to be reduced substantially; still further reduction may be obtained by designing the bearing to give as much hydrodynamic operation as possible, i.e. with as large arcs of bearing surface as possible. A design of bearing based on this approach was recently made by Pametrada. It was a fitted bearing, that is a bearing whose radius was equal to that of the shaft, and clearance was obtained by offsetting the normally unloaded area. The main load-carrying pad was of 90-degree arc, this being the greatest possible length of arc of a fitted bearing to give wholly convergent film conditions, and three shorter pads were provided for stability. Each pad had a generous lead-in at its edges and large axial water grooves were provided, the pad surfaces themselves being plain.

The pads were made of a fabric-reinforced phenolic resin, and it was operated in a test machine which consisted of a gun-metal-lined steel shaft driven through gearing by an electric motor, the shaft and bearing being immersed in a tank of water. The bearing was loaded downwards on to the shaft by the compression of carriage springs, specific loads up to 80 psi (11½ tons dead load) being obtainable at speeds from 1 r.p.m. to 275 r.p.m., and the torque required to drive the shaft could be measured. This bearing was run on sea-water, and at the highest load of 80 psi it was found that preponderantly hydrodynamic operation was obtained at speeds above about 125 r.p.m., this value being lower with lower loads. Apart from operation at lower loads, at 80 psi, specific load 50 hours were completed at 275 r.p.m., 50 hours at 100 r.p.m., and 200 hours at 1 r.p.m. with no significant wear-down. At 110 r.p.m. with full load the bearing showed no appreciable wear-down in 30 hours of operation on a suspension of 8 cwt. of sand in 400 gallons of sea-water, and other tests under misalignment were carried out without trouble.

As a result of this work, the Admiralty decided to fit two such bearings as A-bracket bearings in a cruiser. One of the bearings was made of a fabric reinforced phenolic resin, the other of asbestos-reinforced phenolic resin; their bore was 17⅜ in. and their length 13 in., this being approximately 15 per cent of the length of the conventional bearings they replaced. These bearings have now had a year's evaluation at sea, in which period the ship has steamed about 12,500 miles, made up of 3,100 miles in the range of 50-100 r.p.m., 6,000 miles at 100-150 r.p.m. and 3,400 miles at 150-224 r.p.m. No undue vibration has been experienced, the performance being entirely satisfactory. The shaft liners at the end of this period were found to be in good condition, with no signs of undue wear or scoring. The two bearings were found to be in good condition, one showing slight scoring and the other no damage whatsoever; the wear-down readings were between 0.025 and 0.030 in.

HIGH-SPEED BEARINGS

In most cases the high-speed bearing designer is faced with the problem, with oil-lubrication, of cool-

ing his bearing. With the grades of oil used in high-speed marine machinery the limit of design is frequently being approached, due to this problem, at quite moderate loads. As soon as reliable hydrodynamic lubrication with water is demonstrated, however, the position changes substantially, as indicated earlier, due to the lower viscosity and higher specific heat and density of water as compared with oil, in other words due to the greater cooling capacity of water. Thus at high speeds the losses in a bearing, correctly designed, are lower with water than with oil and the temperature rise is much less.

TURBINE BEARING TESTS

To demonstrate this, a series of tests has been carried out at Pametrada on journal and thrust bearings suitable for a main turbine, the bearing dimensions and conditions of operation being based on a high-speed marine H.P. turbine. They were of 6 in. bore and 0.010 in. diametral clearance, with length/diameter ratios of 0.65 and 0.9, and the materials used were fabric-reinforced or asbestos-reinforced phenolic resins. They were lubricated by unfiltered industrial main fresh water.

The type of test program involved was to run for periods of say 50 hours at 6,000 r.p.m. at stages of load between 200 and 750 psi (specific), examining the bearing and journal at each stage, and then to make 100 stops under load. Finally, water supply was shut off when the bearing was operating at high load and speed and its subsequent performance observed, since the performance of a bearing under failure conditions should also always be examined, because if this leads to disastrous results the bearing design, however satisfactory for normal conditions, is not a fully reliable one.

Typical results were as follows:

Material (a)—fabric-reinforced resin

Operation over the range of load and speed up to 200 psi and 8,000 r.p.m. gave no difficulty; it was observed that the journal did not appear to lift fully until 600-900 r.p.m. had been reached, the actual speed depending on load. No significant wear of bearing or journal was caused in running up to 400 psi or in a large number of stops under load. In the simulated water-supply failure tests the condition of operation was 6,000 r.p.m., 400 psi specific load; the load was held constant as speed was gradually reduced, taking a minute to stop. Examination showed slight charring of the bearing surface. Three further tests of this nature were carried out, in these cases keeping the speed constant, after shutting off the water, until the bearing torque increased rapidly, when the journal rotation was stopped. Further charring was caused, with a maximum bearing wear-down of 0.030 in.; the journal was not damaged and the bearing in each instance operated satisfactorily when the water supply was restored.

Material (b)—asbestos-reinforced resin

This bearing also did not appear to float fully until a speed of about 800 r.p.m. had been reached. In running with normal lubrication up to 750 psi specific load and 8,000 r.p.m. no difficulty was experienced and no significant wear of journal or bearing occurred. The simulated water-failure tests were carried out in the same way as for Material (*a*); the effect on the bearing was similar—considerable charring occurred, with about 0.013 in. wear-down, and the journal surface was slightly damaged. There was no difficulty in running again satisfactorily with the water supply restored.

These results showed that the general design of bearing seemed to be satisfactory giving full hydrodynamic operation at speeds above about 600-900 r.p.m. Below this speed no trouble was experienced, either in steady running or in stopping under load. Under the lubricant-failure condition, extremely satisfactory resistance to damage was experienced.

The results obtained during normal running indicated very low temperature rises, relatively high water flows and flow/speed relationships very different from those obtained with oil-lubricated bearings. The shape of the bearing loss/speed curves suggested that the losses were induced mainly by churning rather than by hydrodynamic shear. The fact that water flow as well as being high was practically independent of bearing speed was to be expected, since the clearances of these water-lubricated bearings were similar to those of equivalent oil-lubricated bearings, in other words the relation between lubricant viscosity and clearance, which has a major effect on flow, was very different for these bearings than for the equivalent oil-lubricated bearings. Similarly, the effect of inlet pressure on water flow was much more marked than the effect with oil, and in other work it has been found that water-lubricated bearings may be run at lower lubricant inlet pressures than when the lubricant is oil, giving considerable further reduction in losses.

THRUST BEARING

In the oil-lubricated thrust bearing for this turbine there are eight pads on each side of the collar at a mean diameter of 7.32 in., running at a maximum speed of 6,600 r.p.m. and specific load of 132 psi, the estimated oil flow being 11 g.p.m. A set of eight such pads was modified by machining off the whitemetal and replacing it by a layer of a type of fabric-reinforced phenolic resin which was available in thin sheets. A suitable adhesive was used to effect the joint and in addition the end pads of each half ring were dowelled. The lubricant used was unfiltered industrial mains fresh water, and this bearing was run in the Pametrada full-scale test machine where it could be loaded by a similar oil-lubricated bearing. The only change in the lubrication arrange-

ments when the bearing was converted from oil to water was to reduce the size of the orifice in the outlet restriction.

The bearing was worked up in speed and specific load to 10,000 r.p.m. and 200 psi, without trouble or dimensional change. A number of 20-hour runs were then carried out at 6,000 r.p.m. and increasing specific load. At and above a load of 500 psi four pads were removed in order to keep the total load required to manageable proportions. No dimensional change or trouble occurred up to and including a run at 700 psi, nor during the following test which consisted of a large number of stops under a load of 700 psi. Finally, at 4,000 r.p.m. the specific load was increased in steps of 100 psi, but after five minutes at 1,400 psi on the water-lubricated bearing it was found that the oil-lubricated bearing, on which the specific load was 700 psi, had failed. The water-lubricated bearing was found to be in perfect condition.

It was found with this bearing that full floating of the pads did not appear to occur until about 800 r.p.m. had been reached, the actual speed depending on load. The temperature rises were very low compared with those of a similar oil-lubricated bearing. At 10,000 r.p.m. for example with 200 psi specific load the water outlet temperature was 95.5° F., or 41.5° F. above inlet; a similar oil-lubricated bearing would have a temperature rise of at least twice this.

The losses of the water-lubricated bearing were greatly affected by lubricant flow, as is also found to be the case with oil-lubricated thrust bearings, but they were lower than with oil, and could be very much lower since lower water flows than oil flows could be used without involving an excessive temperature rise. These factors explain why in the final loading the oil-lubricated bearing failed; since its losses were higher and the cooling capacity of its lubricant lower, oil film break-down due to overheating occurred before any failure in the water-lubricated bearing. The conclusions were that the load and speed for which the oil-lubricated bearing was designed were well within the compass of the water-lubricated bearing. The power loss and temperature rise was less with water than with oil, especially at high speeds, and the water-lubricant bearing withstood a considerably greater overload.

TURBINE BEARINGS

A 4,000 h.p./5,000 r.p.m. turbine which was likely to be in continuous use was modified for water lubrication. The whitemetal-on-gunmetal journal bearings and thrust pads were replaced by bearings similar to those already described. The 300-gallon oil tank and cooler were replaced by a 100-gallon water tank, no cooler being considered necessary, but the existing oil supply and drain pipes for the bearings were retained. The turbine was found to require considerably less nozzle-chest pressure to start after it had been modified, and no trouble was experienced in shutting down, the cooling of the rotor through the bearings being considerably quicker with water than with oil. Frequent bridge-gauge readings have been taken, and over a period of a year and a half in which 1,200 hours' operation have been recorded, no significant dimensional change has been observed. The maximum inlet-outlet water temperature difference has been 10° F. and the operation of the system has been entirely satisfactory.

PRACTICAL IMPLICATIONS OF WATER LUBRICATION

With high-speed bearings lower losses may be achieved than with oil, and the limit of conditions is raised in many cases by the greater cooling capacity of water; with low-speed bearings such as stern-tube bearings much higher loads can be accepted, allowing shorter bearings to be used. In both these applications the requirements which are met by these advantages already exist. With high-speed bearings, for example, there is a continuing requirement for the use of oils of lower and lower viscosity, to give lower losses and greater cooling by virtue of greater flows. With propeller-shaft stern-tube and A-bracket bearings there is a requirement for shorter bearings with the faster-running and lighter naval shafts so that more points of support may be achieved. The use of oils of lower viscosity, however, is often limited by other factors, for instance in marine turbines by the ability of the oil to lubricate the gearing, and the reduction in length of stern-tube bearings of conventional design is limited by that design.

The consequential implications of water-lubrication, however, are in many ways of more importance than the direct results on the bearing. The use of water-lubricated bearings in a water pump, for example, has an effect on sealing arrangements, pump size, and reliability which is probably of much greater value than the effect on bearing efficiency. The inflammability of oil is a serious disadvantage in some cases, and the fact that water may be treated as an expendable lubricant may be an advantage in others.

In the marine-turbine installation immediate possibilities may be seen for the use of water lubrication in auxiliary machinery. For high-speed main machinery the limitation on lowering oil viscosity due to the requirements of lubricating gears, for example, may be overcome, of course, by using a second oil system, as may any incompatibility between other properties of oils, for instance as selected for very high load gears and their operation as a turbine bearing lubricant, but in many cases the complication of a second oil system cannot be countenanced. Water, however, is a fluid for which there will almost invariably be a system for other purposes and therefore it may be used as a lubricant without involving any significant additional equipment or cost.

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BIBLIOGRAPHY OF A DECADE OF RESEARCH ON OIL-ASH CORROSION BY HEAVY FUELS (1948 - 1958)

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Prior to World War II, in which he served approximately 4 years as a naval aviator, he studied mechanical engineering at the University of Iowa. After the war he received a B.S. in mechanical engineering at the University of Maryland. The author is a Lieutenant Commander in the Naval Reserve and is attached to the Reserve Research Company 5-11, Annapolis, Md.

THE USE of heavy or residual type fuels in both gas turbines and steam generating plants has caused severe corrosion problems associated with high temperature service use. Although steam plants in the United States have operated for many years with heavy fuels, no apparent serious slagging or corrosion difficulties were reported until the late 1940's. About this same period the gas turbine in stationary plants was beginning to suffer from corrosion and deposits from heavy fuels.

In 1949 an extensive survey of the petroleum, chemical and power industries was made by a joint ASME-ASTM group to determine to what extent and under what conditions corrosion and slagging of furnaces, boilers and gas turbines had been encountered from the burning of heavy residual oils. It was found that the power industry was experiencing the most severe problems of corrosion and deposits because of the higher operating temperatures, pressures and high ash content fuels being utilized. With the progressive growth and utilization of the more efficient and economical high temperature and high pressure plants serious corrosion problems had arisen in shore and marine installations.

The U. S. Navy, which has always experienced some difficulty with slag formation and corrosion in marine boilers, did not become concerned with these problems until the high temperature-high pressure plants were adopted during World War II. The new operating conditions coupled with restrictive tube spacings in marine boilers, caused formation of a type of slag that did not respond readily to the normal cleaning techniques employed. Although slagging is still a problem, it possibly can be minimized or alleviated by employing various methods of salt water removal from the fuel with concurrent removal of sodium from the fuel. This slagging has been reported as being chiefly sodium sulphate formed from the reaction of the sodium chloride from sea water present in shipboard residual fuels as an emulsion and sulfur contained in the fuel.

Corrosion due to vanadium present in heavy fuels in the forms of oil soluble organo-metallic compounds presents a new and vastly difficult problem. Corrosiveness of these vanadium products on metals at temperatures as low as 1150°F is severe enough to cause a catastrophic oxidation that can consume metallic alloys in a short period of time or reduce their mechanical strength seriously. The precise mechanism of the corrosive attack by this "bad actor" is still not clearly understood. It has been suggested that the attack by vanadium, in the form of the low melting temperature of vanadium pentoxide, takes place by fluxing of the protective metal oxide scale

thus exposing the metal to progressive oxidation. The role of other metallic ash constituents also present in the fuel introduces further complications. The total ash content is generally low, less than 0.05% and although it may be reduced by filtration and centrifuging these processes have no appreciable effect on the vanadium content of the fuel. To date studies have shown that vanadium cannot be removed from the fuel economically so other approaches to deter this active agent are necessary.

The steady utilization of stationary gas turbines as power producers both in the United States and abroad has been chiefly instrumental in promoting a large amount of research aimed toward combating the oil-ash corrosion problems. Economic reasons have caused the gas turbine plants to burn heavy fuels and to operate at high temperatures. These factors have accented the corrosion problems. The growing potential use of gas turbines as marine propulsion equipment, such as main power units installed on the U. S. merchant ships *John Sergeant* and *William Patterson* (the former incorporates a 6000 hp regenerative-cycle gas turbine and the latter a free piston gas producer plant with two 3000 hp gas turbines) and the British Shell tanker *Auris,* has likewise stimulated extensive combustion and corrosion research by private industry and government agencies.

Extensive research both in the laboratory and in the field has been undertaken in the last ten years—ranging from the basic theoretical study of the physical and chemical phenomena of corrosion to the practical application of any expedient means of combating oil-ash corrosion. Super alloys, protective coatings on alloys, combustion control, fuel oil treatment and additives—all have been studied and investigated. Partial relief in specific cases has been reported but no complete answer has yet been found. We have not completely left the experimental stages. The entire problem is still very complex.

The many variables that enter the corrosion picture are all controlling factors that complicate the study of oil-ash deposition and corrosion. For example: in gas turbine applications there are many parameters which influence the corrosion phenomena. To name a few, we have:

1. Metallic ash content of fuel.
2. Temperature of combustion.
3. Pressure of combustion gases.
4. Excess air in combustion gases.
5. Viscosity of fuel at burner.
6. Design of combustion chamber.
7. Fuel atomization.
8. Thermodynamic and aerodynamic conditions.
9. Composition of alloys in blades, nozzles, etc.

Many of these listed factors can apply equally to steam applications since the combustion of residual fuels and the after effects are relatively similar in both applications. However, the gas turbine should experience the more severe condition because of the high inlet gas temperatures and high heat intensities required to produce high specific output for good thermal efficiency.

To date fuel oil treatment and injection of various additives to the fuel or into the furnace and/or combustor have been used with various degrees of success. Several public utility companies have used chemical additives as corrosion inhibitors with good results in both gas turbines and steam generating plants. It has been reported that the additional cost was very small. However, problems of pumping, fuel atomization and deposit formation have become major problems which need further research. The U. S. gas turbine powered merchant ship *John Sergeant* is presently using a combination water washing fuel system with additive injection into the fuel. This system has proved to be very successful with a restricted grade of Bunker C residual fuel. The British Shell Tanker *Auris,* presently being equipped with a 5500 hp gas turbine plant, will also use a similar system for purifying the fuel. Experience with a 1200 hp gas turbine plant aboard this ship confirmed that an open cycle gas turbine burning heavy fuel is a serious rival, in the medium power range, to present conventional types of plants. As time progresses it will be interesting to see the effects on operation and life of these marine gas turbine plants as fuel specification requirements become broader.

This paper lists over 150 technical papers, articles, periodicals, communications, publications, etc. which have been presented at technical societies, printed or published during the past 10 years. Several papers written prior to 1948 are also included because of their basic technical information relative to the over-all subject.

All of the literature in the bibliography are identified by the symbols GT, S, and X. These designations indicate that the material in a respective article is chiefly concerned with the oil-ash problem from:

Gas turbine application (GT)
Steam power application (S)
Experimental approach to corrosion (X)

The latter designation applies to research which has been conducted through basic and applied studies of the over-all oil-ash corrosion problem or a specific approach to a particular phase. This information should therefore be applicable to either gas turbine or steam generated power applications.

It is hoped that this compilation of information will be of aid and value to those researchers who are presently engaged in efforts to study and to help solve the many complex phases of the problems associated with oil-ash corrosion from heavy fuels.

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THE UNITED STATES NAVY

GUARDIAN OF OUR COUNTRY

The United States Navy is responsible for maintaining control of the sea and is a ready force on watch at home and overseas, capable of strong action to preserve the peace or of instant offensive action to win in war.

It is upon the maintenance of this control that our country's glorious future depends; the United States Navy exists to make it so.

WE SERVE WITH HONOR

Tradition, valor, and victory are the Navy's heritage from the past. To these may be added dedication, discipline, and vigilance as the watchwords of the present and the future.

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THE FUTURE OF THE NAVY

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Mobility, surprise, dispersal, and offensive power are the keynotes of the new Navy. The roots of the Navy lie in a strong belief in the future, in continued dedication to our tasks, and in reflection on our heritage from the past.

Never have our opportunities and our responsibilities been greater.

Obituary

Marcus Herndon Goshen

On June 12, 1958, Marcus Herndon Goshen, one of the Society's oldest and most valued members, died at Vallejo, California, at the age of 80.

Mr. Goshen was born at Petaluma, California, on May 3, 1878. He began his career in shipbuilding as a draftsman in the Union Iron Works, completing his work there in 1904. He then entered Government Service in the office of Inspector of Machinery, Moran Bros. Shipyard, Seattle, Washington as an estimator and became Chief Estimator. In this capacity he directed preparation of Machinery Plans for the battleship NEBRASKA and on completion of the ship in 1907 he was transferred to Mare Island Naval Shipyard. In 1910, when the yard first undertook to organize "Central Planning" he was selected to head the Civilian Planning & Estimating Force and in 1913 was made Chief Planner and Estimator. He remained in this capacity until his retirement in 1944.

Mr. Goshen joined the Society on June 29, 1904.

He is survived by his widow.

We regret to announce the death of the following:

Eaton, Philip B., Naval Member
Fisher, Thomas N., Naval Member
Scott, Ralph W., Civil Member

BOOK REVIEW

"THE COMPLETE BOOK OF SUBMARINES"
By Commander C. W. Rush, USN, W. C. Chamblin, and H. J. Gimpel
Published in 1958 by
The World Publishing Company, 2231 West 110th Street, Cleveland 2, Ohio
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Reviewed by
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4. Anatomy of a Submarine
5. Eyes and Ears
6. Submarine Weapons
7. The Enemy Above
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BOOKS RECEIVED

"Single Sideband for the Radio Amateur," Second Edition, 1958; edited by the Headquarters Staff of the American Radio Relay League, West Hartford 7, Connecticut; 262 pages, $1.50 (U. S.), $1.75 (elsewhere).

"Handbuch fur Schiffsingenieure und Seemaschinisten," edited by E. Ludwig and K. Illies, published in 1958 by Friedr. Vieweg and Sohn, Braunschweig, Germany. Written in German.

"Danger in the Air," by Oliver Stewart, published in 1958 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N. Y.; 194 pages, $6.00.

"The Structure of Steel," by Dr. Edwin Gregory and Eric N. Simons, published in 1958 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N. Y.; 176 pages, $10.00.

"Recent Research in Maritime Transportation" (The proceedings of the second symposium convened by The Maritime Cargo Transportation Conference), published in 1958 by The National Academy of Sciences—National Research Council; Publication No. 592; 134 pages; paper backed; $2.00.

"Advances in Steel Technology in 1956," prepared by The Secretariat of the Economic Commission for Europe, United Nations, Geneva, published in 1958 and distributed by Columbia University Press, 2960 Broadway, New York 27, N. Y.; paper backed, mimeographed; $1.00.

"Effect of Surface on the Behavior of Metals" (Lectures delivered at the Institution of Metallurgists Refresher Course, 1957), published in 1958 by Philosophical Library, Inc., 15 East 40th Street, New York 16, N. Y.; 100 pages including 30 diagrams in the text and 10 pages of art plates; $10.00.

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